THE POWER HANDBOOKS

ARITHMETIC

OF THE

STEAM BOILER

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A REFERENCE BOOK
SHOWING THE VARIOUS APPLICATIONS OF
ARITHMETIC TO STEAM BOILERS

BY
CHARLES J. MASON

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THIS BOOK

IS RESPECTFULLY DEDICATED

TO

MR. FRED R. LOW, EDITOR OF POWER,

WHO HAS EVER TAKEN A DEEP INTEREST IN THE

AFFAIRS OF THE ENGINEERING FRATERNITY

PREFACE

This book is a compilation of arithmetical rules and formulas applicable to steam boilers of various types. The author claims no originality in the preparation of the material, excepting only the arrangement and manner of presentation.

It is intended as a book of reference for those who may require rules and formulas directly related to steam boilers, and its aim is concentration and logical order in the arrangement and treatment of the various features introduced.

Most of the material was gathered during the author's career as a steam and marine engineer, covering a period of twenty-five years.

It is not intended to teach the elements and principles of arithmetic in this book, as might perhaps be inferred from the title, but only the application of arithmetic to steam-boiler calculations. It is presumed that those who may use it already understand arithmetic but desire to have a compact set of rules and formulas conveniently ready for use, without having to look through several books for a certain one when required. Those who are preparing for examinations for engineer's certificates and licenses will find the work of great assistance to them.

The author desires to thank all those who have in any way contributed to the production of this work,

particularly the publishers and editor of *Power*, from which paper various extracts have been taken, kind permission to use the same having been granted, and Mr. William Kent, M. E., author of Kent's Mechanical Engineer's Pocket-book.

CHARLES J. MASON.

SCRANTON, PENNA,

December, 1913.

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BOILER CALCULATIONS

CHAPTER I

The Sphere; Stress Per Square Inch Section; Safe Pressure

The sphere is the strongest form in which a steam boiler could be made, but because of mechanical and commercial reasons, that form is not used. In order to understand the stresses, due to pressure, endured by steam boilers, it is well to start with the spherical form, for that is the simplest, and it forms a basis for calculations on the prevailing forms in which boilers are made.

Given a spherical vessel made of metal of a certain thickness and known diameter, it is desired to find what stress per square inch section the metal is subjected to, due to a known pressure per square inch contained within the sphere. The pressure would tend to separate the sphere in halves, through a diametral plane. Actually, pressure in a closed vessel of any form radiates from the center outward. But for convenience in calculations the radiating forces may be resolved in two, and acting perpendicular to any diametral plane. The total pressure or force tending to burst the sphere asunder will be the product of the known pressure per square inch,

existing in the sphere, and the area in square inches of a diametral plane.

For illustration, assume a pressure of 100 lb. per square inch; a sphere whose internal diameter is 30 in., and made

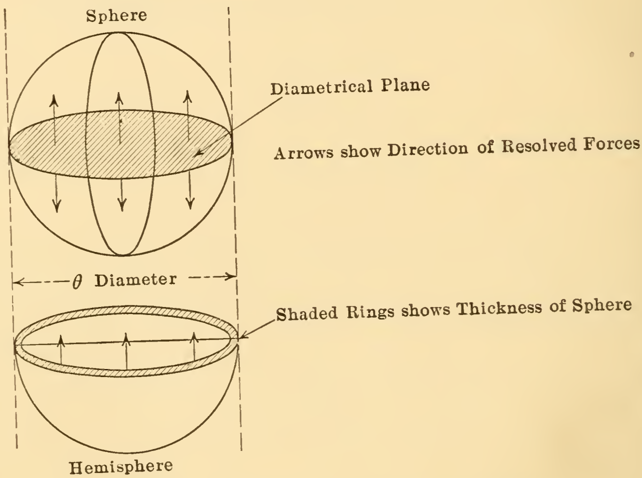

FIG. 1.—Diagram of stresses in a sphere.

of 1/2-in. steel plate, no joints, seams, nor rivets. The area of the diametral plane is:

$$30^2 \times .7854 = 706.860 \text{ sq. in.}$$

As the pressure is 100 lb. on each square inch, then:

$$100 \times 706.860 = 70686.00 \text{ lb.}$$

This is the total force tending to separate the sphere in two parts.

This force is resisted by the area of metal at the cir-

cumference of the plane upon which the pressure is assumed to act. This is actually a ring whose internal diameter is 30 in. and whose outside diameter is 31 in. This is called *sectional area*, and it is the difference between the area of the inner and outer circles of which the ring of metal is formed.

The area of a circle whose diameter is 31 in. is:

$$31^2 \times .7854 = 754.769 \text{ sq. in.}$$

and,

$$30^2 \times .7854 = 706.860 \text{ sq. in.}$$

$$\text{Difference} = 47.909 \text{ sq. in.}$$

That is, 47.909 sq. in. cross-sectional area of steel plate is enduring a total stress of 70,686 lb. or,

$$\frac{70686}{47.909} = 1475.42 \text{ lb.}$$

per square inch section.

Assuming the tensile strength of the plate to be 60,000 lb. per square inch section, the foregoing shows that considerably more than 100 lb. per square inch could safely be carried in the sphere considered, for with a factor of safety of 6, the stress would be 10,000 lb. and this would be obtained by having the pressure at 677.77 lb. per square inch, as a trial in calculation will show.

Ordinarily, in practical work, the difference between the outside and inside diameters is not taken.

For example, the rule that would be employed to find the stress per square inch in a sphere whose inside diameter is 30 in., and whose thickness of plate is .5 in., is: Multiply the area of the diametral plane by the pressure per square inch, and divide that product by the product of the inside diameter, the constant 3.1416,

and the thickness of the plate in inches. This written in formula is:

$$\frac{d^2 \times .7854 \times p}{d \times 3.1416 \times t} = \text{stress.}$$

But by simple cancellation the formula reduces to

$$\frac{p \times d}{4 \times t} = \text{stress.}$$

The value of the letters used is:

$d =$ inside diameter, inches;
$p =$ pressure, pounds per square inch;
$t =$ thickness of plate, inches;
$s =$ safe stress in pounds per square inch;
$.7854 =$ a constant;
$3.1416 =$ a constant.

In order to find the pressure per square inch that may safely be carried, when the safe stress, the diameter of sphere, and the thickness of plate are given, it is simply a matter of changing the rule and formula to suit the purpose, thus:

Multiply four times the thickness of plate by the given safe stress per square inch section, and divide by the internal diameter in inches.

Written in formula it is:

$$\frac{4t \times \text{stress}}{d} = p.$$

Applying this to the example chosen, it becomes:

$$\frac{4 \times .5 \times 10000}{30} = 666.66 \text{ lb.}$$

which for practical purposes is 667 lb. per square inch. Here, the internal diameter only has been taken. In the first method shown the mean diameter was taken, which gave a safe pressure of 677.77 lb. per square inch which for practical purposes is 678 lb. The difference is 11.1 lb. or 1.6+ per cent. difference in favor of the usual method, which, though not absolutely correct, errs on the side of safety as shown.

If the sphere in the example were .25 in. thick, or 60 in. internal diameter, then the difference in the methods explained would be less than that given.

Grouping all the rules and formulas pertaining to the sphere, so that any term may be found having the remaining terms given:

To find the stress per square inch section endured by the plates, multiply together the pressure per square inch and the internal diameter in inches. Divide the product by four times the thickness of the plate in inches.

Written as a formula this rule becomes:

$$\frac{p \times d}{4 \times t} = s \qquad (1)$$

To find the safe pressure per square inch that may be carried, multiply four times the thickness of the plate by one-sixth of the ultimate tensile strength of the material of which the plates are made, and divide the product by the internal diameter in inches.

$$\frac{4 \times t \times s}{d} = p \qquad (2)$$

To find the internal diameter, multiply four times the thickness of the plate by one-sixth of the ultimate tensile strength of the

material of which the plates are made, and divide by the safe pressure per square inch.

$$\frac{4 \times t \times s}{p} = d. \qquad (3)$$

To find the thickness of plate, multiply the internal diameter by the safe pressure per square inch, and divide the product by one-sixth the ultimate tensile strength of the material of which the plates are made; divide the quotient by 4.

$$\frac{p \times d}{4 \times s} = t. \qquad (4)$$

The Cylinder

Next to the sphere, the cylinder is the form best suited for steam boilers, and excepting the spherical form, the

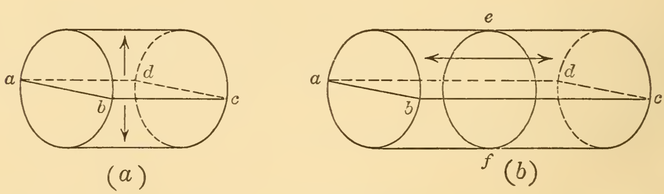

FIG. 2.—Stresses in cylinders. (*a*) Stress acting at right angles to the longitudinal plane *a b c d*. Tendency to separate the cylinder at *a d* and *b c*. (*b*) Stress acting parallel to the longitudinal plane *a b c d*. Tendency to separate the cylinder at *e f*.

cylindrical is the strongest. This is not hard to understand, when it is considered that pressure inside any vessel tends to make that vessel assume a spherical form, as illustrated in the inflated toy rubber balloon. For this reason flat surfaces in boilers must be braced, but cylindrical surfaces do not require any bracing.

For the sake of convenience in calculations, the force due to the pressure in a cylindrical vessel may be considered as acting in two directions. One acts in the direction tending to blow off the head, and the other at right angles to the diametral longitudinal plane. In the former, the calculations are exactly the same as has been explained in connection with the sphere. In the latter, the pressure acting against the imaginary plane tends to separate the cylinder into two parts, and that which resists the tendency is the area of metal along both sides of the cylinder.

The stress per square inch section of the plate, due to any given pressure, may be found from the following rule:

Multiply together the pressure per square inch, the diameter of the cylinder in inches, and the length in inches; divide the product by two times the thickness of the plate in inches multiplied by the length in inches.

This, written as a formula, is:

$$\frac{p \times d \times l}{2 \times t \times l} = s. \tag{5}$$

As the factor l appears both above and below the line, it may be cancelled, and the formula becomes,

$$\frac{p \times d}{2 \times t} = s. \tag{6}$$

From the formula just given it can be seen that the stress increases as the diameter or pressure increases, and it also can be seen that as the thickness of the plate increases the stress on the plates decreases.

By comparing formula (1) with formula (6) it is seen

that in the latter the stress is twice as great as it is in the former. A careful study of these rules and formulas will show why the stress in one case is just twice that of the other. For that reason, the longitudinal seams in a boiler must be made stronger than the circumferential seams, and this is accomplished by having more than one row of rivets in the longitudinal seams.

Circumferential seams are frequently double riveted to make a mechanically tight job. The strength of riveted joints will be treated further on. So far, seamless vessels have been considered, so as to introduce the subject of stress and resistance to stress in the most simple form.

The following rules relate to the cylindrical form, and belong in the group under consideration.

To find the stress on each inch in the circumference tending to blow off the head in a *longitudinal* direction (not the stress per square inch section), the following rule is applicable:

Multiply together the area in square inches of the end of the cylinder and the pressure per square inch, and divide the product by the circumference of the cylinder in inches.

$$\frac{d^2 \times .7854 \times p}{d \times 3.1416} = s. \tag{7}$$

To find the total stress caused by the pressure in a cylinder, multiply together the diameter in inches, the length in inches, and the pressure in pounds per square inch.

$$d \times l \times p = \text{total } s \tag{8}$$

l = length in inches.

To find the total pressure on the entire shell of a cylinder, multi-

ply together the circumference in inches, the length in inches, and the pressure per square inch.

$$c \times l \times p = \text{total pressure.} \qquad (9)$$

To find the bursting pressure of a cylinder, multiply together the thickness of the plate in inches and the tensile strength of the material of which the plate is made, and divide the product by the radius of the cylinder in inches.

$$\frac{t \times T}{r} = \text{bursting pressure per square inch.} \qquad (10)$$

In this formula, t = thickness of plate in inches. T = tensile strength in pounds per square inch section of the material, and r = the radius in inches.

To find the safe working pressure of a cylinder, multiply together the thickness in inches and the tensile strength of the material of which the plate is made, and divide that product by the radius in inches multiplied by whatever factor of safety may be desired.

$$\frac{t \times T}{r \times f} = \text{safe working pressure per square inch.} \qquad (11)$$

Here follows examples showing the application of the foregoing rules and formulas from (1) to (11) inclusive. For the sake of clearness and convenience the same values will be used in all. Numbers easy to operate have been chosen, for no matter what numbers may be contained in any example which may come up in practice, the *method of operation* will be exactly the same.

In formula (1) assume the following values:

pressure (p) = 100 lb. per square inch;
diameter (d) = 60 in.;
thickness (t) = 1/2 or .5 in.

Then to find the stress (s) the statement becomes:

$$\frac{100 \times 60}{4 \times .5} = 3000 \text{ lb. per square inch section.}$$

To find the safe pressure (p) which may be carried, formula (2), the statement becomes:

$$\frac{4 \times .5 \times 3000}{60} = 100 \text{ lb. per square inch.}$$

To find the internal diameter (d), formula (3), the statement becomes:

$$\frac{4 \times .5 \times 3000}{100} = 60 \text{ in.}$$

To find the *thickness of plate* (t), formula (4), the statement becomes:

$$\frac{100 \times 60}{4 \times 3000} = .5 \text{ in.}$$

Formula (5) reduces to that given in (6), and to find the *stress per square inch section* in this case, the statement becomes:

$$\frac{100 \times 60}{2 \times .5} = 6000 \text{ lb.}$$

per square inch section.

To find the *stress* (s) *on each inch in the circumference*, formula (7), the statement becomes:

$$\frac{60^2 \times .7854 \times 100}{60 \times 3.1416} = 1500 \text{ lb.}$$

To find the *total stress* on the entire shell, formula (8), assume a length of 144 in. with the other values remaining the same, the statement becomes:

$$60 \times 144 \times 100 = 864{,}000 \text{ lb.}$$

To find the *total pressure* on the entire shell, formula (9), the statement becomes:

$$60 \times 3.1416 \times 144 \times 100 = 2,714,342.4 \text{ lb.}$$

To find the *bursting pressure*, formula (10), the statement becomes: assume a tensile strength of 50,000 lb. per square inch section.

$$\frac{.5 \times 50000}{30} = 833.33+ \text{ lb. per square inch.}$$

To find the *safe working pressure*, formula (11), and assume a factor of safety of 5, the statement becomes:

$$\frac{.5 \times 50000}{30 \times 5} = 166.66+ \text{ lb.}$$

In practice 170 lb. would be allowed with factor of safety 5 per square inch. Or a simpler method is to take one-fifth of the bursting pressure thus:

$$\frac{833.33}{5} = 166.66 \text{ lb.}$$

and as the bursting pressure is five times the safe pressure,

$$5 \times 166.66+ = 833.33+ \text{ lb.}$$

In actual practice, 6 is used as a factor of safety more frequently than 5. However, the *method* of operation is the same for any and all factors of safety that may be used; so that if the method is understood, it matters not as to the values that may be substituted in the various formulas treated of.

Riveted Joints

In the previous sections, spheres and cylinders without joints or seams were assumed in order to simplify explana-

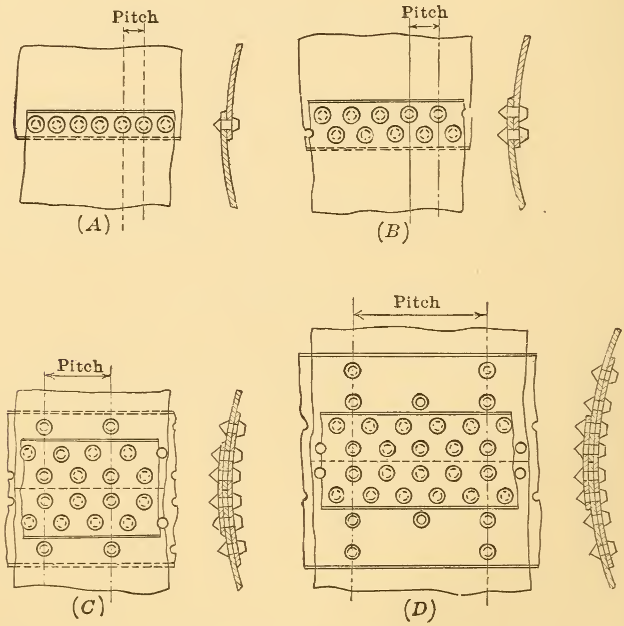

Fig. 3.—Types of riveted joints. (*A*) Single-riveted lap joint; one rivet in single shear. (*B*) Double-riveted lap joint; two rivets in single shear. (*C*) Triple-riveted, double butt-strapped joint; four rivets in double shear, one rivet in single shear. (*D*) Quadruple-riveted, double butt-strapped joint; eight rivets in double shear, three in single shear.

tions and calculations. In actual practice, however, steam boilers are constructed with both longitudinal

and circumferential joints, or seams. The seams are secured by rivets. There are several kinds of riveted joints known to engineers and others who have to do with boilers. The strength of a riveted joint depends upon how the joint is made, as to the size and pitch or spacing of the rivets, and the number of rows of rivets, and also as to whether the joint is what is known as a *lap* or *butt-strapped* one. A riveted joint of any kind is not theoretically as strong as the solid part of the plate, although in practice it has been known for boilers to tear apart at some place other than at the riveted joint. This probably was due to a flaw or weakness in the metal. If a cylindrical vessel were made of plates uniform in structure and thickness throughout, and if tested to destruction, it would likely break or pull apart at the riveted longitudinal joint. In making calculations it is presumed that a break would occur at the joint, rather than at any other place.

Strength of Riveted Joints

The strength of a riveted joint is compared with that of the solid plate, the latter being valued at 100 per cent. Nominally, the strength of joints varies from 56 per cent. to 94 per cent., the former value representing single-riveted lap joints, and the latter, quadruple, double butt-strapped joints. Between single-riveted lap joints and quadruple-riveted, double butt-strapped joints, there are: double-riveted lap joints, triple-riveted lap joints, quadruple-riveted lap joints, single-riveted butt joints, double-riveted butt joints, triple-riveted butt joints. The foregoing joints may be either chain riveted,

or what is termed zig-zag riveted. In triple-riveted joints and in quadruple-riveted joints it is customary to omit every alternate rivet in the outer rows; this admits of a stronger joint.

The strength of a riveted joint depends upon the size and pitch of the rivets, the number of rows of rivets, type of joint as to lap or butt, the tensile strength of the plates, and the shearing stress of the material of which the rivets are made. In finding the strength of a joint two things are to be considered, the strength of the plate section and the strength of the rivet section. The lesser value, as found from an analysis, is taken as the strength of the joint as a whole.

Theoretically, riveted lap joints and those butt joints with one cover plate should be designed so that the rivet and plate sections are equal—or as nearly equal as possible—in strength. But in practice it is usually considered desirable to so design a joint that the plate section is a little stronger than the rivet section; this particularly relates to joints having *all* the rivets either in single or double shear, for the reason that the plates become thin from wear, with a consequent reduction in strength, while the rivets suffer little if any from wear. But in joints having some of the rivets in single shear and some in double shear, the greatest strength usually obtains when the rivet section exceeds the strength of the net section of plate.

Efficiency of Riveted Joints

To determine the efficiency of a riveted joint, its resistance must be calculated for each of the different ways

in which it may fail, and then the lowest efficiency so found in relation to the solid plate, will be the one by which the joint is known. A riveted joint may fail in the following ways:

(1) The plate may break asunder along the rivet holes, at the net section.

(2) The rivets may shear off, leaving the plates intact.

(3) The plate may shear out in front of the rivets.

(4) The plate may crush in front of the rivets.

(5) In joints having zig-zag rivets the plate may break diagonally between the rivet holes.

(6) The joint may fail by a combination of the foregoing. With joints as usually proportioned, the liability to failure in the ways referred to in (3), (4), (5) and (6) is reduced to a minimum; this by having the distance from the *edge* of the plates to center of rivet holes one and one-half times the diameter of the rivet holes. It is customary (except in special cases which will be referred to) to consider only (1) and (2) as possible ways of failure, and base all calculations upon those two ways. Therefore, calculate the efficiency of the net section of the plate part of the joint as compared with the solid plate, and then find the efficiency of the rivet section of the joint as compared with the solid plate; the lesser of the values found is to be taken as the final efficiency of the joint as a whole.

Single-riveted Lap Joints

To find the efficiency of a single-riveted lap joint. (The distance from the edge of the plate to the center

line of rivet holes must be not less than one and one-half times the diameter of the rivet hole, for all joints.)

First, find the strength of a unit of length of the solid plate.

$$P \times t \times S = \text{strength of solid plate.}$$

In which,

P = pitch of rivets in inches, from center to center.
t = thickness of plate in inches.
S = tensile strength of plate, in pounds per square inch section.
d = diameter in inches of rivet holes.

The next step is to find the strength of the net section of plate between the rivet holes:

$$(P-d) \times t \times S = \text{strength of plate between the holes.}$$

Next, find the shearing strength of one rivet in single shear:

$$n \times s \times a = \text{shearing strength of one rivet.}$$

n = number of rivets in single shear; s = shearing strength of rivet; a = cross-section area of rivet, after driving.

Take the lesser of these two results and divide it by the value found for the strength of the solid strip; the quotient will be the efficiency of the joint decimally expressed.

Example.—Single-riveted lap joint of the following dimensions:

S = 55,000 lb.
P = 1.625 in.
t = .25 in.
d = .6875 in.
s = 42,000 lb.
a = .3712 sq. in.

The strength of the solid strip will be:

$$1.625 \times .25 \times 55{,}000 = 22{,}343 \text{ lb.}$$

The strength of the net section of plate between the rivet holes will be:

$$(1.625 - .6875) \times .25 \times 55{,}000 = 12{,}890 \text{ lb.}$$

The strength of the rivet in single shear will be:

$$1 \times 42{,}000 \times .3712 = 15{,}590 \text{ lb.}$$

As the net section of plate in this example is weaker than the rivet, its value must be used. Then:

$$\frac{12890}{22343} = .576, \text{ or } 57.6 \text{ per cent. efficiency.}$$

Shearing Strength of Rivets

The shearing strength of rivets may be taken from the following table (from the Massachusetts Board of Boiler Rules):

Iron rivets in single shear,	38,000 lb.
Iron rivets in double shear,	70,000 lb.
Steel rivets in single shear,	42,000 lb.
Steel rivets in double shear,	78,000 lb.

These values are on the safe side, as they are lower than some others that are in use.

Double-riveted Lap Joints

To find the efficiency of double-riveted lap joints, the method of procedure is the same as that for single-riveted joints, with the exception that there are *two* rivets in single shear instead of *one* as in single-riveted joints.

Example.—A double-riveted lap joint has the following dimensions:

S = 55,000 lb.
t = .3125 (5/16) in.
P = 2.875 (2 7/8) in.
d = .75 (3/4) in.
a = .4418 sq. in.
s = 42,000 lb.

The strength of the solid plate is:

$$2.875 \times .3125 \times 55{,}000 = 49{,}414 \text{ lb.}$$

The strength of the net section of plate is:

$$(2.875 - .75) \times .3125 \times 55{,}000 = 36{,}523 \text{ lb.}$$

The strength of the two rivets in single shear is:

$$2 \times 42{,}000 \times .4418 = 37{,}111 \text{ lb.}$$

Here again the plate section is the weaker, so the value for that must be used:

$$\frac{36523}{49414} = .739, \text{ or } 73.9 \text{ per cent. efficiency of joint.}$$

Triple- and Quadruple-riveted Lap Joints

In triple-riveted and quadruple-riveted lap joints (sometimes used in marine boilers) there are *three* and *four* rivets, respectively, in single shear. With this exception, the method of finding the efficiency of such joints is the same as for single and double, as just illustrated.

Lap Joints

Lap joints for the longitudinal seams are now considered not safe for steam boilers of more than 36 in. in

diameter, and for pressures higher than 100 lb. per square inch; and probably as time goes on they will not be used at all; but it is important to know how to calculate the strength of such joints, hence the reference to them in this book.

Butt Joints

Butt joints with double cover plates are the strongest and safest joints in use. The *minimum* thickness of *cover plates*, or *butt straps* as otherwise called, is as follows:

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Thickness of shell plates	Minimum thickness of butt straps
1/4 in.	1/4 in.
5/16 in.	1/4 in.
3/8 in.	5/16 in.
7/16 in.	3/8 in.
1/2 in.	7/16 in.
9/16 in.	7/16 in.
5/8 in.	1/2 in.
3/4 in.	1/2 in.
7/8 in.	5/8 in.
1 in.	3/4 in.
1 1/8 in.	3/4 in.
1 1/4 in.	7/8 in.

Single Butt Straps

Single butt straps should never be thinner than the plates of the shell. In some instances (British Board of Trade and Canadian Rules) the minimum thickness must be not less than one and one-eighth the thickness of the shell plates. Double butt straps must be at least

five-eighths, and preferably the thickness of the shell plates. If the shell plate is light, say 7/16 in. or less, the outside strap should be as heavy as the plate, to admit of a tightly calked joint.

When *single* butt straps are used, the method of finding the efficiency of the joint is the same as that for lap joints, for the rivets are all in single shear, and the pitch of the rivets is the same in each row.

Number of Rivets Considered

In lap joints, *all* the rivets in a given pitch strip of plate are taken into account when figuring for the efficiency of joint, while in butt joints *only those rivets* on *one side* of the center line of the joint are considered. A little thought on the part of the reader will make clear the reason.

Number of Rows of Rivets

In double butt-strapped joints, three or four rows of rivets on each side of the center line are generally used. In the former (triple riveted) the pitch of the *outer* row of rivets on *each side* of the center line is *twice* the pitch distance of the two inner rows of rivets on each side of the center line.

In the latter (quadruple-riveted joints) the *outer row* of rivets on each side of the center line of joint is *four* times the pitch distance of the two inner rows on each side of the center line; in the rows next to the outer rows, the rivets are pitched *twice* the distance of those in the two inner rows. (See Fig. 3.)

Number of Rivets in Double and Single Shear

In triple-riveted butt joints, there are *four* rivets in double shear, and *one* rivet in single shear, in a given pitch strip.

In quadruple-riveted butt joints, there are *eight* rivets in double shear, and *three* rivets in single shear, in a given pitch strip.

When calculating the efficiency of triple- and quadruple-riveted joints, the strength of the net section of plate is taken at the *outer* row of rivets, where the pitch is the greatest. The reason for this will be explained presently.

High Joint Efficiencies Due to Wide Spacing of Rivets at the Outer Rows

It is because of the wide spacing in the outer rows of rivets that such high efficiencies can be obtained with those types of joints as compared with those joints in which the pitch of the rivets is the same for all the rows.

In order to explain *why* the net section of the plate at the outer row of rivets is taken, an illustrative example of a triple-riveted, double butt-strapped joint will be used; the same principles may be applied to a quadruple joint of the same kind.

Thickness of shell plates 3/8 in., tensile strength 50,000 lb. per square inch section, rivet holes 13/16 in., rivets 3/4 in. diameter; shearing stress of the rivets taken as 38,000 lb. per square inch section. The pitch of rivets in the two inner rows is 3 1/4 in., and in the outer row, 6 1/2 in.

The width of strip to be considered in this case is 6 1/2 in. The sectional area is $.375 \times 6.5 = 2.4375$ sq. in. $2.4375 \times 50,000 = 121,875$ lb. strength of the solid strip, with which the joint is to be compared.

Next find the strength of the net section of the plate at the outer row of rivets.

As there is but 1 rivet in the 6 1/2 in. strip under con-

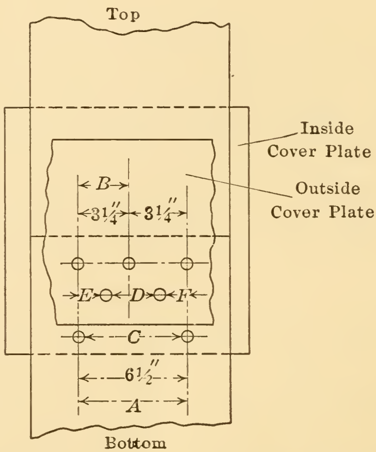


FIG. 4.—Diagram of triple-riveted double butt-strapped joint.

sideration, at the outer row, then, $6.5 - .8125 = 5.6875$ in. width of net section of plate; and, $5.6875 \times .375 \times 50,000 = 106,640$ lb. strength of net section of plate between the rivet holes.

If the plate should break or pull asunder at the line of the outer row of rivets, the resistance to breaking is the metal in the net section of the plate as shown at *c* in Fig. 4. But should the plate break along the net

section at the inner row of rivets, the resistance offered is the strength of the sections of plate *E D* and *F*, and as well as that, the resistance to shearing offered by one-half of each of the rivets in the outer row, which is equivalent to one rivet in calculation.

First, find what the plate resistance is. By measurement it is $6.5-(2\times.8125)=4.875$ in. Here there are *two* rivet holes to subtract from the width of the strip. The sectional area will be $4.875\times.375=1.828$ sq. in.; and $1.828\times50{,}000=91{,}400$ lb. resistance.

To this must be added the resistance offered by the two half rivets in the outer row. The area of a 13/16-in. driven rivet is .5185 sq. in., and as the shearing stress is 38,000 lb. per square inch, $.5185\times38{,}000=19{,}703$ lb. This added to 91,400 lb. before obtained for the plate gives 111,103 lb. total resistance to the plate breaking at the inner row of rivets, as against the 106,640 lb. found for the net section of plate at the outer row of rivets. Therefore the latter is the weaker, and there the plate will probably break, if at all.

Continuing, there are four rivets in double shear, and one rivet in single shear, in the strip. The resistance to shearing of one rivet was shown to be 19,703 lb. The resistance to shearing offered by *each* rivet in *double* shear is:

$$38{,}000\times.85=32{,}300 \quad\text{and}\quad 38{,}000+32{,}300=70{,}300 \text{ lb. per square inch section.}$$

(The factor .85 is a value used for rivets in double shear as double shear does not necessarily mean twice that of single shear.) The area of each rivet is .5185

sq. in., and there are four to consider, therefore, .5185 $\times 4 \times 70{,}300 = 145{,}802$ lb. and to this add the value for the rivet in single shear, 19,703 lb., which gives a total of 165,505 lb. shearing strength of all the rivets in the strip. The net section of plate at the outer row of rivets is the weakest part of the joint as a whole, and its value is to be compared with the strength of the solid strip in order to find the efficiency. The net section of plate is 106,640.625 lb. and the solid strip is 121,875 lb.; the efficiency is:

$$\frac{106640.625}{121875} = 87.5 \text{ per cent.}$$

Quadruple-riveted, Double Butt-strapped Joint

To analyze a quadruple-riveted joint and find the efficiency, proceed as follows:

Fig. 5 shows the construction and arrangement of rivets. The data is given in this manner. A strip of the joint marked P is taken. The value of the letters is:

P = pitch of rivets in inches.
t = thickness of plate in inches.
S = tensile strength of plates.
d = diameter of the driven rivets, in inches.
N = number of rivets in double shear.
n = number of rivets in single shear.
a = area of cross-section of rivets, in square inches.

Strength of the solid strip of plate considered = $P \times t \times S$, represented by letter A.

Strength of plate between the rivet holes at the *outer* row of rivets $= (P - d) \times S$, represented by the letter B.

The shearing strength of 8 rivets in double shear, plus the shearing strength of 3 rivets in single shear $=Na+na$, represented by the letter C.

The strength of the plate between the rivet holes in the *second* row plus the shearing strength of 1 rivet in single shear in the *outer* row $=(P-2d)\times t\times S+na$, represented by the letter D.

Next, divide B, C, or D, whichever is the least in value,

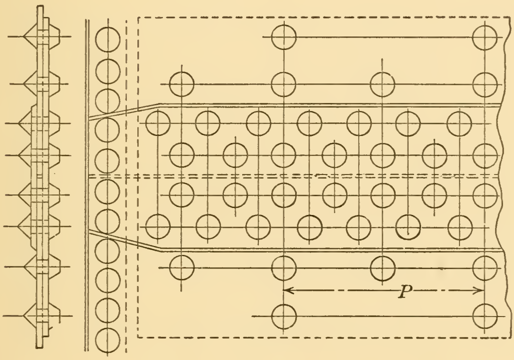


FIG. 5.—Quadruple-riveted double butt-strapped joint.

by the value of A, and the quotient will be the efficiency of the joint.

The numerical values to be used are:

S =55,000 lb.
t =1/2 in. or .5.
P =15 in.
d =15/16 in. or .9375 in.
a =.6903 sq. in.
N =8 rivets, double shear value of 78,000 lb.
n =3 rivets, single shear value of 42,000 lb.

The final values are:

$A = 15 \times .5 \times 55{,}000 = 412{,}500.$

$B = (15 - .9375) \times .5 \times 55{,}000 = 386{,}718.$

$C = 8 \times 78{,}000 \times .6903 + 3 \times 42{,}000 \times .6903 = 517{,}725.$

$D = (15 - 2 \times .9375) \times .5 \times 55{,}000 + 1 \times 42{,}000 \times .6903 = 389{,}930.$

The value of B is found to be the least, therefore the strength of the joint depends upon the weakest part, and the efficiency is,

$$\frac{386718}{412500} = .937 \text{ or } 93.7 \text{ per cent.}$$

The foregoing example shows how to calculate the efficiency of the various parts of the joint where possible failure may occur. The same line of reasoning, and similar methods of operation, may be employed for *any kind or type of riveted joint* that may be used in a steam boiler.

Size of Rivets and Pitch

There are no absolute rules for determining the size of rivets to be used in any given case, for with different size rivets, the same efficiency of joint may be obtained. The size to be chosen depends upon several factors and varies with any one of them. The things to be considered are: the shearing strength of the material of which the rivets are made, the tensile strength of the plates to be riveted together, the pitch of the rivets, and the type of joint to be made. The required efficiency of joint determines the type to be used and the greatest pitch of rivets allowable depends upon the thickness of plate to be used, the object sought being steam-tight joints.

For single-riveted lap joints, the United States Supervising Inspectors of Steam Vessels recommend a rivet diameter equal to the plate thickness plus 7/16 in. using steel plates and steel rivets. For double-riveted lap joints, a rivet diameter equal to the plate thickness plus 3/8 in. Some authorities make the rivet diameter range from plate thickness plus 3/8 in. to plate thickness plus 1/2 in., with plates from 1/4 in. to 1/2 in. thickness.

For triple-riveted lap joints rivet diameters range from plate thickness plus 3/8 in. to plate thickness plus 7/16 in. with plates from 1/4 to 1/2 in. thickness.

For double-riveted butt joints, triple-riveted butt joints, and for quadruple-riveted butt joints, rivet diameters range from plate thickness plus 5/16 in. to plate thickness plus 7/16 in. The foregoing is intended to give a general idea only of rivet sizes that may be chosen to be somewhat in proportion to the joint as a whole. In the end it is a matter of choosing that size rivet and a certain pitch which will give the highest efficiency of joint, consistent with steam-tight work, type of joint, strength of materials and all other considerations. It is a matter of "cut and try" until the best is arrived at.

Distance Between Adjacent Rows of Rivets

The distance between adjacent rows of rivets, center to center, is sometimes called *transverse pitch.* When the rivets are subjected to the same kind of shear, this distance should not be *less* than *twice* the diameter of the rivets, nor *more* than *two and one-half times* the diameter of the rivets used. If the distance between the rows of

rivets is too small, the plate is likely to fracture along a diagonal line, or diagonal pitch as it is termed.

If the transverse pitch is at least equal to twice the diameter of rivets, failure of the plate will not occur along the diagonal line, but rather in the net section of plate along the line of rivets. This makes the calculation of joint efficiency somewhat simpler. In cases where the outer butt strap is not as wide as the inner strap, the distance between the line of rivets in double shear and the line in single shear should be two and three-quarters or three times the diameter of rivets, in order to have a properly formed rivet head, and also room to calk the outer strap.

Safe Working Pressure of Cylinders with Riveted Joints

It will be remembered that formula 11, page 9, gives the safe working pressure of a cylinder without any visible joint. But cylinders having riveted joints must be calculated with the efficiency of longitudinal joint taken into consideration. The rule will be the same as that expressed in formula 11, with the additional factor of joint efficiency expressed as a decimal value.

Let e represent the efficiency of the longitudinal seam or joint; the efficiency of the girth seam is not required for reasons explained at the beginning of this chapter.

The formula now becomes:

$$\frac{t \times T \times e}{r \times f} = \text{safe working pressure per square inch.}$$

Using the same example illustrating formula 11, page 11, and assuming a joint efficiency of say 85 per cent. the statement becomes:

$$\frac{.5 \times 50000 \times .85}{30 \times 5} = 141.66 \text{ lb. safe pressure per square}$$

inch. No matter what the efficiency of the joint may be, nor by what method it may be found, it is always to be applied as shown in the example just given. Of course there are other ways of arranging and simplifying the factors in the formula, but no matter what the arrangement or how simplified, the result will always come out the same if the work is correctly done. Abbreviated formulas and rules are convenient to those who know of their derivation, but they are not satisfying to those who do not know just how each simplified factor was obtained. For this reason, no attempt is made in this work to abbreviate anything that will detract from the value of any problem presented, as far as underlying elements and principles are concerned. Any one who understands how to calculate the efficiency of riveted joints, and how to find the safe working pressure of spherical or cylindrical vessels as given in this work, will be able to work out similar problems, no matter in what form they may be given, or what rule it may be desired to apply to them.

So far, the shell and its riveted joints only have been considered. The rules given apply to the cylindrical parts of all boilers of whatever make. There are other rules relating to shells of special design, and these rules will be given further on.

The next in order, at present, is the bracing or staying of boiler heads and flat surfaces in boilers.

CHAPTER II

Boiler Heads—Unstayed Heads

Bumped heads may be either convex or concave according as to how placed in a shell. Fig. 6 shows the application of the two forms, (*a*) being a concave bumped head while (*b*) is a convex head. The arrows show the direction of pressure acting against the heads. Bumped heads do not require bracing, particularly the convex (*b*) form

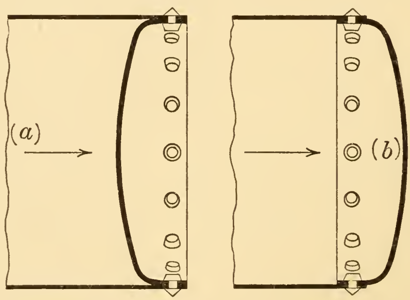


Fig. 6.—Bumped heads. (*a*) Concave head. (*b*) Convex head. Arrows show the direction of pressure.

as it is already in the form that internal pressure would tend to make it assume. In the concave (*a*) form, the tendency of internal pressure is to collapse the head, and allowance is made for this in the rule which will be given presently for the safe working pressure allowed. Bumped

heads may be either single or double riveted to the shell.

It is necessary to know the radius to which a head is bumped when making calculations for safe working pressure. A bumped head is presumed to be virtually part of the surface of a sphere. To find the radius to which a head is bumped, take half the diameter of the head where it fits into the shell, and multiply that value by itself, and divide the product by the height of the

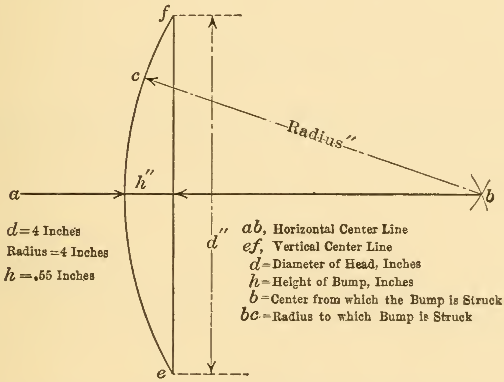


FIG. 7.—Diagram showing how to find radius of bumped head.

bump. To the quotient add the height of the bump and divide the sum by 2. It is usual to take the dimensions in inches.

To find the radius to which a boiler head is bumped the following formula may be used:

Referring to Fig. (7),

$$\frac{\dfrac{(\frac{1}{2}\times d)^2}{h}+h}{2}=\text{radius.}$$

All dimensions to be taken in inches.

To find the safe working pressure of a bumped head like that in (*b*) and when it is *single riveted* to the shell, the following formula may be used:

$$p=\frac{t\times S}{3\times r} \tag{1}$$

When the head is *double riveted* to the shell, then the formula becomes:

$$p=\frac{t\times S}{2.5\times r} \tag{2}$$

When the head is concaved like that in (*a*) and *single riveted*, then the formula becomes:

$$p=\frac{t\times S}{5\times r}. \tag{3}$$

When a concaved head is *double-riveted* the formula becomes:

$$p=\frac{t\times S}{4\frac{1}{6}\times r}. \tag{4}$$

In these formulas the values of the letters are:

p = safe working pressure in pounds per square inch.
t = thickness of metal in the head, expressed in inches.
r = radius in inches, to which the head is formed.
S = tensile strength of the material of which the head is made, expressed in pounds per square inch section.

As the foregoing formulas are similar in construction, one example will serve to illustrate the operation of all.

A boiler head is bumped to a radius of 60 in. made of plate .5 in. thick, with a tensile strength of 50,000 lb. and *double riveted* to the shell. What working pressure will be allowed?

The operation is as follows:

$$p = \frac{.5 \times 50000}{2.5 \times 60} = 166.66 \text{ lb. per square inch.}$$

Unstayed Flat Heads

When a boiler head is flat and not stayed, the following formula may be used:

$$p = \frac{t \times S}{.54 \times A}$$

The letters have the same values as for the preceding formulas, and A equals the area of the head expressed in square inches.

Example.—A flat head is 30 in. in diameter, .75 in. thickness of plate having a tensile strength of 55,000 lb. per square inch, what pressure will be allowed?

Operation:

$$p = \frac{.75 \times 55000}{.54 \times 30 \times 30 \times .7854} = 108 \text{ lb. per square inch.}$$

All the rules for boiler heads of the kind just described are those prescribed by the Board of Supervising Inspectors of Steam Vessels in the United States.

If it is desired to find what thickness a bumped head should be, having the tensile strength, the radius to

which the head is bumped, and the pressure in pounds per square inch to be carried, it is a matter of transposing the terms of the proper formula in the group just treated of.

Suppose in the last illustrative example given that it is desired to know what thickness of head should be employed. The formula transposed will be:

$$\frac{p \times 2.5 \times r}{S} = t$$

Applying this to the example, the statement becomes:

$$\frac{166.66 \times 2.5 \times 60}{50000} = .5 \text{ in. thickness.}$$

Thickness of Boiler Heads, Massachusetts Rules

In actual practice, however, the thickness of boiler heads is not derived mathematically but empirically. The rules in the state of Massachusetts require the thickness to be as follows: Boilers up to and including 42 in. diameter, heads must be 3/8 in. From 42 in. to 54 in. diameter, heads must be 7/16 in. From 54 in. to 72 in. diameter, heads must be 1/2 in. Over 72 in. diameter, heads must be 9/16 in.

Thickness of Boiler Heads, Ohio Rules

The rules formulated for bumped heads by the Board of Boiler Rules in the State of Ohio differ slightly from those given.

The *minimum* thickness of a *convex* head shall be determined by this formula:

$$\frac{R \times F.S. \times P}{T.S.} = t$$

The *minimum* thickness of a *concave* head shall be determined by this formula:

$$\frac{R \times F.S. \times P}{.6(T.S.)} = t$$

In these two formulas the values are as follows:

R = one-half the radius to which the head is bumped.
$F.S.$ = 5 = factor of safety.
P = working pressure, in pounds per square inch, for which the boiler is designed.
$T.S.$ = tensile strength, in pounds per square inch, stamped on the head by the manufacturer.
t = thickness of head in inches.

The *radius* of head shall not exceed the *diameter* of the shell.

When a convex or concave head has a manhole opening, the thickness as found by the formulas just given, shall be increased by not less than 1/8 in.

The minimum thickness of plates in stayed flat surface construction shall be 5/16 in.

The *minimum* thickness of tube sheets shall be as follows:

"When the diameter of tube sheet is 42 in. or less, the thickness is 3/8 in.; over 42 in. to 54 in. inclusive, 7/16

in.; over 54 in. to 72 in. inclusive, 1/2 in.; over 72 in., 9/16 in.

Stays and Stay Bolts

The maximum allowable stress per square inch net cross-sectional areas of stays and stay bolts as defined in the Massachusetts rules, is as follows:

Weldless, mild steel head to head or through stays, 8000 lb. for sizes up to and including 1 1/4 in. diameter, or equivalent area, and 9000 lb. for sizes over 1 1/4 in. diameter or equivalent area. Fig. 8 (*a*) illustrates direct or through stay arrangement.

Weldless, mild steel diagonal or crow-foot stays, 7500 lb. for sizes up to and including 1 1/4 in. diameter, or equivalent area and 8000 lb. for sizes over 1 1/4 in. diameter or equivalent area.

Weldless, wrought-iron, head to head or through stays, 7000 lb. for sizes up to and including 1 1/4 in. in diameter or equivalent area, and 7500 lb. for sizes over 1 1/4 in. diameter or equivalent area.

Weldless, wrought-iron, diagonal or crow-foot stays, 6500 lb. for sizes up to and including 1 1/4 in. or equivalent area, and 7000 lb. for sizes over 1 1/4 in. diameter or equivalent area.

Welded mild steel or wrought-iron stays, 6000 lb.

Mild steel or wrought-iron stay bolts 6500 lb. for sizes up to and including 1 1/4 in. diameter or equivalent area, and 7000 lb. for sizes over 1 1/4 in. diameter or equivalent area.

When a greater allowable stress per square inch on

stays and stay bolts is required than those just given, the material shall conform to the following physical qualities:

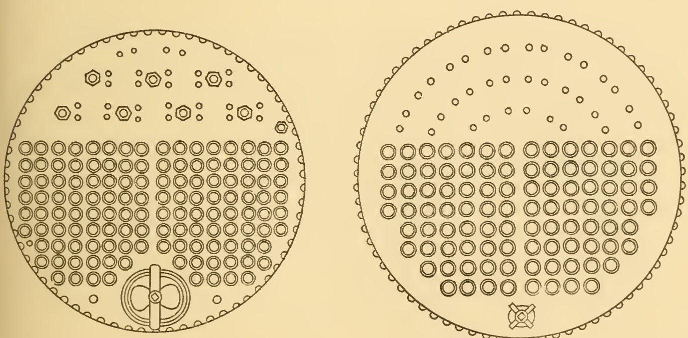
The tensile strength shall not exceed 62,000 lb. per square inch.

The yield point shall not be less than one-half the tensile strength.

The elongation per cent. in 8 in. shall not be less than 28.

Direct Stays

To find the *safe working pressure* per square inch that may be carried by stays of a given size, the following formula may be applied:



(a) (b)

Fig. 8.—Arrangement of stays in boiler heads. (*a*) Direct through stays in horizontal rows, seven stays. (*b*) Diagonal stays in concentric rows, seventeen stays.

$$\frac{a \times S}{A} = p \qquad (1)$$

To find the *diameter* of stays required,

$$\frac{A \times p}{S} = a, \text{ and } \sqrt{\frac{a}{.7854}} = d \qquad (2)$$

To find the *area* supported by one stay, and the *distance between stays*,

$$\frac{A \times S}{p} = A, \text{ and } \sqrt{A} = D. \qquad (3)$$

To find the required tensile stress that may be endured by stays:

$$\frac{A \times p}{a} = S \qquad (4)$$

The value of the letters in this group of rules is as follows:

A = area in square inches, supported by one stay.
a = area, cross-sectional, in square inches, of stays.
p = pounds pressure per square inch.
S = tensile strength of stays in pounds per square inch.
d = diameter of stays in inches.
D = distance between stays, in inches.

Example, illustrating the foregoing rules.

Assume stays of 1 in. diameter at the smallest part, with an allowable stress of 6000 lb. per square inch, and distanced 6 in. center to center.

The area of each stay will be $1^2 \times .7854 = .7854$ sq. in.

The area supported by each stay will be $6 \times 6 = 36$ sq. in.

Applying these values to formula (1) the statement becomes:

$$\frac{.7854 \times 6000}{36} = 130.9 \text{ lb. per square inch allowable pressure.}$$

To formula (2),

$$\frac{36 \times 130.9}{6000} = .7854 \text{ sq. in. area of each stay.}$$

and, $$\sqrt{\frac{.7854}{.7854}} = 1 \text{ in. diameter of stays.}$$

To formula (3),

$$\frac{.7854 \times 6000}{130.9} = 36 \text{ sq. in. area supported by each stay.}$$

and, $$\sqrt{36} = 6 \text{ in. distance center to center of stays.}$$

To formula (4),

$$\frac{36 \times 130.9}{.7854} = 6000 \text{ lb. tensile stress allowed on stays.}$$

In the foregoing no allowance has been made for the space occupied by the stays in the sheets supported. This is on the side of safety and is generally accepted. If in any case it is not accepted, it becomes a matter of subtracting the area occupied by the stays from the area as found from the center to center measurement.

Diagonal Stays

The size of a diagonal stay depends upon the angle it makes with the surface it is helping to support, when considered in relation to a direct stay. The less the angle is, the larger in diameter must the stay be. The nearer a stay is to being at right angles to the surface it supports, the smaller in diameter it may be. The same principle

applies to any form of stay that may be used, other than a circular cross-section.

In Fig. 9 is shown an ordinary diagonal stay attached to the boiler head at C and to the shell at E. The length of the stay is considered as CE; the distance CD also enters the calculation as will be shown presently.

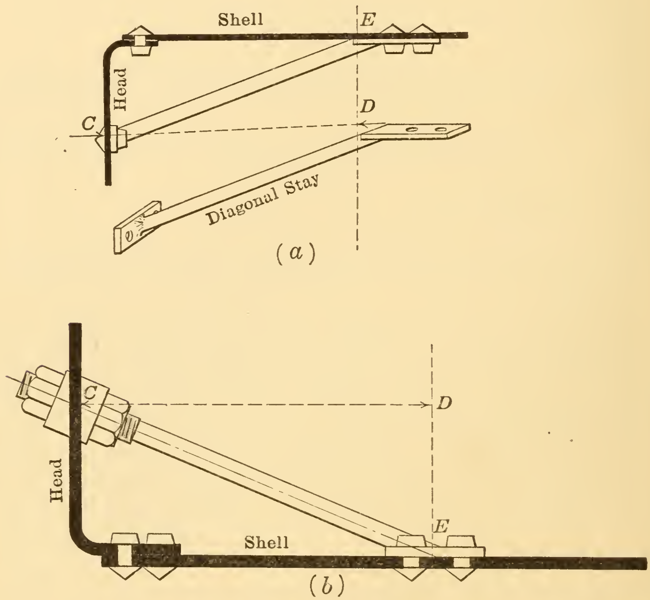


FIG. 9.—Forms of attachment of diagonal stays. (*a*) Riveted at both ends. (*b*) Riveted at one end, nuts and washers at the other end.

To find the area of a required diagonal stay, first find the area of a direct through stay, as has been explained. Call the length CE of the required diagonal stay, x and the distance, CD, y.

Let a = the area of direct stay.
Let A = the area of diagonal stay.
Let S = tensile strength of stay.

Then,

$$\frac{a \times x}{y} = A \qquad (1)$$

$$\frac{A \times y}{a} = x \qquad (2)$$

$$\frac{A \times y}{x} = a \qquad (3)$$

$$\frac{a \times x}{A} = y \qquad (4)$$

$$\frac{\frac{A \times y}{x} \times S}{\text{area supported by one stay}} = \text{Safe pressure per square inch that may be carried.} \qquad (5)$$

Example, illustrating the foregoing group of rules.

Assume that the area of a direct stay has been found to be .7854 sq. in. (due to 1 in. diameter as before taken). That the length of the required diagonal stay is to be 36 in. and that the distance between perpendiculars of points of attachment is 30 in. Then, applying these values to formula (1),

$\frac{.7854 \times 36}{30} = .9425$ sq. in., nearly, cross-sectional area of diagonal stay required, and the corresponding diameter is:

$$\sqrt{\frac{.9425}{.7854}} = 1.095 \text{ in. diameter.}$$

The nearest commercial size brace that would be used is 1 1/8 in. diameter.

To formula (2),

$\frac{.9425 \times 30}{.7854} = 36$ in., length of the required diagonal stay, as measured on the line CE in Fig. 9.

To formula (3),

$\frac{.9425 \times 30}{36} = .7854$ sq. in., area of direct or through stay.

To formula (4),

$\frac{.7854 \times 36}{.9425-} = 30$ in., distance between perpendiculars as measured on the line CD in Fig. 9.

To formula (5),

$\frac{\frac{.9425 \times 30}{36} \times 6000}{49} = 96.1$ lb. per square inch pressure allowed.

Other formulas for obtaining the diameters of direct and diagonal stays.

For direct stays,

$$d = \sqrt{\frac{A \times p}{S \times .7854}} \tag{1}$$

For diagonal stays,

$$d = \sqrt{\frac{x \times A \times p}{y \times S \times .7854}} \tag{2}$$

The values of the letters are the same as those used for the stay group of formulas.

Examples, illustrating these formulas.

Assume 36 sq. in. area supported by one stay (pitch, 6 in. center

to center), pressure to be carried, 100 lb. per square inch, 6000 lb. allowed stress per square inch on stays, then to find the diameter of direct stay as stated in formula (1) the statement becomes:

$$d=\sqrt{\frac{36\times 100}{6000\times .7854}}=.8735 \text{ in.}; \tag{1}$$

in practice 7/8-in. diameter stays would be used.

For formula (2) diagonal stays, assume same values, and in addition, that the length of the diagonal stays are 36 in., and that the horizontal distance between the ends of stays—as shown in Fig. 9—is 30 in., the statement becomes:

$$d=\sqrt{\frac{36\times 36\times 100}{30\times 6000\times .7854}}=.957 \text{ in.}; \tag{2}$$

in practice 1-in. diameter stays would be used in this case.

The area in square inches supported by one stay multiplied by the number of pounds pressure per square inch carried in any given case gives the load in pounds the stay must sustain.

The cross-sectional area in square inches of each stay in any group, multiplied by the allowable tensile stress per square inch section, gives the allowable load in pounds the stay may sustain with safety.

The load, in pounds, on a stay divided by the cross-sectional area of the stay in square inches will give the stress in pounds per square inch.

To find the number of stays of a given size required to support a given area, multiply the area of the sheet to be stayed by the steam pressure in pounds per square inch to be carried, and divide the product by the total allowable stress for the given size of stay.

Example.—Assume 1 1/8-in. round iron stay bolts with an allowable stress of 6000 lb. per square inch, and an area of 2000 sq. in. to be supported against a pressure of 90 lb. per square inch. How many stays are required?

$$(1\ 1/8)^2 \times .7854 \times 6000 = 5964 \text{ lb.}$$

allowable on each stay. Then:

$$\frac{2000 \times 90}{5964} = 30.18, \text{ say 31 stays will be required.}$$

Rivets Securing Stays

The combined cross-sectional area of the rivets which secure the stays to boiler heads must not be less than that of the stay itself. The rivets securing a diagonal stay to the head are in tension chiefly, but they also endure a certain amount of bending. Those rivets which secure the end of the stay to the shell are in single shear chiefly, and to a certain extent are in tension. In practical work the rivets used for both ends of such stays are of the same size, a high factor of safety being used to make due allowance for the different stresses, and the difference in value of rivets in tension and in shear. It is considered advisable to allow but 4000 lb. per square inch section for rivets used on diagonal stays, in order to be on the safe side.

The part of a head above the tubes to be braced is a segment of a circle. In laying out the position of the stays it will be found that they cannot be arranged so that exactly the same load will be borne by all. It is customary to arrange the stays in concentric rows, as shown in Fig. 8 (*b*).

It is usual to consider that the flange of a boiler head supports the head for a distance of at least 3 in., measured from the inside of the flange. With modern methods of making plates and flanging them, the radius to which a head is flanged is now greater in proportion to the thickness than used to obtain, and it is thought by some that more than 3 in. may be counted upon as being supported by the flange, depending on the thickness of the head, the pressure to be carried, and the disposition of the stays. Speaking in general, the 3-in. distance is reasonable and safe to use.

The tubes that are expanded into the tube sheet support that part of the head. A certain portion of the head above the top row of tubes is supported by them, depending on the size and holding power of the tubes themselves.

A reasonable and safe distance to consider in such calculations is that of one-half of the bridge between the tubes. Suppose, for example, that in a given boiler the tubes are 3 1/2 in., spaced 4 3/4 in. center to center; the bridge, or section of plate between the tubes is 1 1/4 in.; then one-half that distance or 5/8 in. may be considered as being supported above the top row of tubes by the tubes in the top row.

Diagonal braces of the crowfoot type usually have two rivets spaced 4 in. center to center. This permits of a proper spacing of stays, which is the main point to consider in laying out a head. The maximum allowable pitch must not be exceeded. The braces can be made to suit the load. Formula (7) decides just what pitch may be used.

Flat Surfaces to be Stayed, in which the Thickness of Plate Enters the Calculation

$$A=\frac{112\times t^2}{p},$$ for plates up to 7/16 in. thick. (1)

$$A=\frac{120\times t^2}{p},$$ for plates above 7/16 in. thick. (2)

$$A=\frac{140\times t^2}{p},$$ for screw stay bolts and nuts. (3)

$$p=\frac{112\times t^2}{A},$$ for plates 7/16 in. and under. (4)

$$p=\frac{120\times t^2}{A},$$ for plates over 7/16 in. thick. (5)

$$p=\frac{140\times t^2}{A},$$ for screw stay bolts and nuts. (6)

$$S=\sqrt{\frac{112\times t^2}{p}},$$ for plates 7/16 in. thick and less. (7)

$$S=\sqrt{\frac{120\times t^2}{p}},$$ for plates above 7/16 in. thick. (8)

$$S=\sqrt{\frac{140\times t^2}{p}},$$ for screw stay bolts and nuts. (9)

$$t=\sqrt{\frac{p\times A}{112}} \tag{10}$$

If this formula gives more than 7/16 in. for the value of t, use the next formula with the factor 120 in it.

$$t=\sqrt{\frac{p\times A}{120}} \tag{11}$$

$$t=\sqrt{\frac{p\times A}{140}} \tag{12}$$

In the foregoing group of formulas, the values are as follows:

A = area in square inches, supported by one stay. The pitch can be found by extracting the square root of A.

t = thickness of plate, expressed in sixteenths of an inch. Example: if the plate were 7/16 in. thick, t in this case would be 7. In other words, t is the numerator of the fraction whose denominator is 16.

p = pressure in pounds per square inch, allowed to be carried.

S = pitch of stays, center to center.

112, 120, 140 are constants used in the respective formulas.

Examples, illustrating the application of the formulas in this group:

Assume plate 7/16 in. thick; pressure to be carried, 100 lb. per square inch. Required, area that may be supported by one stay, from which the pitch distance may be found.

Then, in formula (1),

$$A=\frac{112\times 7^2}{100}=54.88 \text{ square inches supported by one stay.}$$

In formula (4),

$$p=\frac{112\times 7^2}{54.88}=100 \text{ lb. per square inch pressure allowed.}$$

In formula (7),

$$S=\sqrt{\frac{112\times 7^2}{100}}=7.4081 \text{ in. center to center of stays.}$$

In formula (10),

$$t=\sqrt{\frac{100\times 54.88}{112}}=7,$$ numerator of the fraction 7/16, thickness of plate in inches.

Formulas (4), (5), and (6) in this group serve as a check on the design of stays, for by applying the proper one to any given case it can be determined as to whether or not the desired pressure may be carried.

Segments to be Braced

In Fig. 10 is shown that portion (shaded) of a boiler head that requires to be braced. The flange is presumed

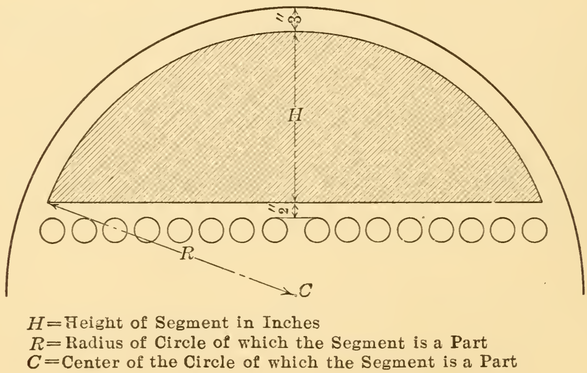


Fig. 10.—Segment of head to be braced.

to support the head for a distance of 3 in., and the tubes, a distance of 2 in. above as shown in the figure. These are the generally accepted figures by most designers of boilers and experience seems to have proved them to be safe.

To find the height of the segment requiring bracing, subtract 5 in. from the distance from the tubes to the highest part of the shell. To find the diameter of the circle of which the segment is a part, subtract 6 in. from the diameter of the boiler.

To find the area of a segment sufficiently close for all practical purposes, the following formula may be used (see Fig. 10):

$$\frac{4 \times H^2}{3} \times \sqrt{\frac{2 \times R}{H} - .608} = \text{area.}$$

In which H = the height of segment, R = the radius of circle, and the numerals, constants.

Suppose, for example, it is desired to know the area of a segment of a circle whose diameter is 50 in., the height of the segment being 20 in. The statement becomes:

$$\frac{4 \times 20^2}{3} \times \sqrt{\frac{2 \times 25}{20} - .608} =$$

$$\frac{1600}{3} \times \sqrt{\frac{50}{20} - .608} =$$

$$\frac{1600}{3} \times \sqrt{2.5 - .608} =$$

$$\frac{1600}{3} \times \sqrt{1.892} =$$

$$\frac{1600}{3} \times 1.375$$

$$\frac{1600 \times 1.375}{3} = 733.3 \text{ sq. in. area.}$$

The formula just given, when applied to segments whose heights are three-tenths of the diameter of the circle of which the segment is a part, gives results correct within 1/10 of 1 per cent.

Method of Finding the Area of a Segment by Use of a Table

As well as the formula that has been given and illustrated by an example, the area of a segment may be found by using the table on page 51, and by an approximation method illustrated in Fig. 11.

It is desired to find the area of the shaded segment of the circle in Fig. 11, by using the table. The height of the segment is 16 in., and the diameter of the circle of which the segment is a part is 54 in. The method of operation is as follows: $\frac{16}{54} = .2963$, which is the height of a similar segment of a circle whose diameter is 1.0. The two nearest values, *Ht.* column in the table, to .2963 are .29 and .3. The corresponding areas are, for .29 = .18905; for .3 = .19817.

A convenient value to use without going into interpolation is .2963 = .195, and $54^2 \times .195 = 568.620$ sq. in. area.

As a check on the foregoing, the formula may be applied to the same example:

$$\frac{4 \times 16^2}{3} \times \sqrt{\frac{54}{16} - .608} = 567.780 \text{ sq. in. area.}$$

AREAS OF CIRCULAR SEGMENTS

Ht.	Area	Ht.	Area	Ht.	Area	Ht.	Area	Ht.	Area
0.01	0.00133	0.11	0.04701	0.21	0.1199	0.31	0.20738	0.41	0.30319
0.02	0.00375	0.12	0.5338	0.22	0.12811	0.32	0.21667	0.42	0.31304
0.03	0.00687	0.13	0.06	0.23	0.13646	0.33	0.22603	0.43	0.32293
0.04	0.01054	0.14	0.6683	0.24	0.14494	0.34	0.23547	0.44	0.33284
0.05	0.01468	0.15	0.7387	0.25	0.15355	0.35	0.24498	0.45	0.34278
0.06	0.01924	0.16	0.08111	0.26	0.16226	0.36	0 25455	0.46	0.35274
0.07	0.02417	0.17	0.08854	0.27	0.17109	0.37	0.26418	0.47	0.36272
0.08	0.02943	0.18	0.09613	0.28	0.18002	0.38	0.27386	0.48	0.3727
0.09	0.03501	0.19	0.10390	0.29	0.18905	0.39	0 28359	0.49	0.3827
0.1	0.04087	0.2	0.11182	0.3	0.19817	0.4	0.29337	0.5	0.3927

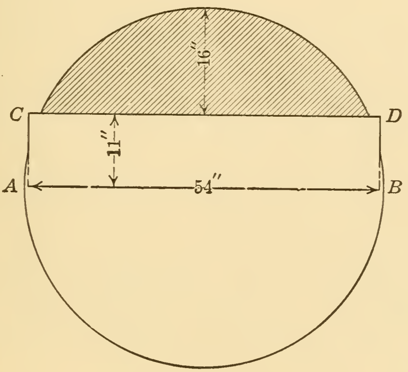


FIG. 11.—Approximate method of finding the area of segment of a circle. Area of *A B C D*, $11 \times 53 = 583$ square inches. Area of semicircle, $\dfrac{54^2 \times .7854}{2} = 1145.115$ square inches. Then, $1145.115 - 583 = 562.115$ square inches, area of shaded segment.

By the table,	568.620
By the formula,	567.780
Difference,	.840 sq. in.

$$\text{Difference per cent.} = \frac{.840}{567.780} \times 100 = .148$$

which is sufficiently close for all practical purposes in boiler work.

The table gives the areas of segments from a height of .01 up to .5 (a semicircle) increasing by hundredths. The areas of segments are from a circle of unit diameter. If the values in the table are taken as feet, they apply to a circle 1 ft. in diameter; if the values are taken in inches, they apply to a circle 1 in. in diameter.

An approximate method is illustrated also in Fig. 11. Consider the length of the rectangle *ABDC* as 1 in. less than the diameter of the circle, and the width as 11 in. This gives an area of $53 \times 11 = 583$ sq. in. This is to be subtracted from the area of the semicircle of which the segment is a part.

The area of a 54-in. circle is $54^2 \times .7854 = 2290.23$ sq. in., and of the semicircle, $\frac{2290.23}{2} = 1145.115$ sq. in. From this subtract 583, which gives 562.115 sq. in. as the area of the segment by this method. Comparing this answer with that obtained by the formula:

$$567.78 - 562.115 = 5.665 \text{ sq. in. difference.}$$

The difference per cent. is:

$$\frac{5.665}{567.78} \times 100 = .99,$$

or 1 per cent. in round numbers, for this particular case. It must not be thought that the approximate method just described will *always* differ from that of the formula by 1 per cent. The difference may vary as much as 2 per cent., according as to the size of the circle and the height of the segment as ordinarily found in stationary

boiler practice. In circles having a diameter from 30 in. to 102 in., the result found by the approximate method will be in error less than 2 per cent., when the height of the segment is not less than three-tenths the diameter of the circle. The Hartford Steam Boiler Inspection and Insurance Company sanction the use of the approximate method, because of its simplicity, and because it gives results sufficiently close for all practical purposes in relation to segments of horizontal tubular boiler heads to be braced.

Girder Bars

Girder bars are of two kinds, the split bar, Fig. 12 (*a*), and the solid bar (*b*). The figure shows the application

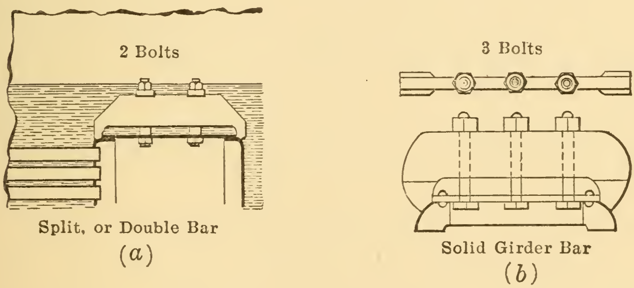


Fig. 12.—Girder bars.

of girder bars to the combustion chamber tops of marine boilers and to the crown sheets of locomotive boilers.

The safe working pressure for solid girders, Fig. 12 (*b*), may be found from the following formula:

$$P = \frac{C \times d^2 \times t}{(w - p) \times D \times l},$$

in which the value of the letters are:

$P=$ safe working pressure in pounds per square inch.
$d=$ depth of the girder in inches.
$t=$ the thickness of girder in inches.
$w=$ width in inches of combustion chamber, measured in direction at right angles to the length of the bar.
$p=$ pitch of bolts in inches, securing bars to the sheet.
$D=$ distance in inches, from center to center of girders.
$l=$ length in feet, of girders.
$C=$ a constant, according to design, as follows: 550 for bars with one bolt, 825 for bars with two or three bolts, 935 for bars with four bolts.

The safe working pressure when split girder bars, Fig. 12 (*a*), are used, may be found from this formula:

$$P=\frac{24000\times t\times d^2}{D\times l^2}$$

In which the value of the letters are:

$P=$ safe pressure in pounds per square inch.
$t=$ thickness in inches of one girder.
$d=$ depth in inches of one girder.
$D=$ distance in inches, from center to center of girders.
$l=$ length in inches of girders.
$24{,}000=$ a constant.

The following example shows the application of the first formula.

A solid rectangular steel girder bar 3 ft. long, 8 in. deep, 2 in. thick, three bolts, bars spaced 8 in. center to center, pitch of bolts 8 in., and combustion chamber top 32 in. wide. What pressure may be carried?

In this example the value of C is 825. Then:

$$P = \frac{825 \times 8^2 \times 2}{(32-8) \times 8 \times 3} = \frac{105600}{576} = 183.3 \text{ lb. per square inch}$$

The values quoted for C are those prescribed by the United States Board of Supervising Inspectors of Steam Vessels.

The following example shows the application of the second formula.

A split or double girder bar, 8 in. deep, 1 in. thick, 30 in. long, the bars being placed 8 in. center to center. What pressure may be carried?

$$P = \frac{24000 \times 1 \times 8^2}{8 \times 30^2} = \frac{1536000}{7200} = 213.3 \text{ lb. per square inch.}$$

The strength of stay bolts, and the load carried by each one fitted to girder bars, are calculated in the same way as for any flat surface; and the same rules apply for the maximum pitch and safe working pressure as for any flat surface.

CHAPTER III

Manhole Reenforcing Rings

Reenforcing rings are for the purpose of strengthening manhole openings in boilers. When only one ring is used it may be placed either inside or outside of the shell. When two rings are used, one is placed inside and the other outside the shell.

· Single rings may be from 1 1/4 to 1 1/2 times the thickness of the shell plates, and when double rings are used the thickness of each may be the same as that of the shell plates. The rings must be riveted to the shell by a sufficient number of rivets of proper size, so that the combined resistance to shearing will be at least equal to the resistance of the rings to tensile stress. The various formulas for reenforcing rings are as follows:

$W=\frac{L\times t_1}{2\times t}+D.$ (1) For single-riveted rings.

$W=\frac{L\times t_1}{2\times t}+2\times D.$ (2) For double-riveted rings.

Formulas (1) and (2) are for *single* rings.

$W=\frac{L\times t_1}{4\times t}+D.$ (3) For single-riveted rings.

$W=\frac{L\times t_1}{4\times t}+2\times D.$ (4) For double-riveted rings.

Formulas (3) and (4) are for *double* rings.

The values of the letters are as follows:

$W=$ width of rings in inches.
$t=$ thickness of rings in inches.
$t_1=$ thickness of shell plates in inches.
$D=$ diameter of driven rivet in inches.
$L=$ length of opening in shell in inches.

In calculating the least number of rivets to be employed, the net section of the ring is to be used. In single riveting, the net section is found by subtracting the diameter of the rivet hole from the width of the ring, and then by multiplying the remainder by the thickness of the ring. When the ring is double riveted, subtract *twice* the diameter of the rivet holes from the width of the ring, and multiply the remainder by the thickness. The diameter of rivet holes is the equivalent of the driven size of rivets, and refers to the trial size of rivets to be chosen in the following formulas:

$$N=\frac{4\times T\times A}{S\times(D^2\times.7854)} \qquad (5) \text{ For } \textit{single} \text{ rings.}$$

$$N=\frac{8\times T\times A}{1.85\times S\times(D^2\times.7854)} \qquad (6) \textit{ Double} \text{ rings.}$$

The values of the letters are:

$N=$ number of rivets required.
$A=$ net section of ring in square inches.
$T=$ tensile strength of ring per square inch section.
$S=$ shearing strength of rivet per square inch section.
$D=$ trial diameter of driven rivet in inches.
$.7854=$ a constant used in finding circular areas.

4 = a constant used for *single* reenforcing rings.

8 = a constant used for *double* reenforcing rings.

If either of these formulas give a too small number of rivets, too widely spaced, a smaller trial diameter of rivet must be chosen and the formula again applied. If on the other hand too many rivets are found, causing them to be too close together, a larger trial diameter of rivet must be chosen.

Examples showing the application of the foregoing group of formulas.

A manhole in a boiler is 11×15 in.; the 11-in. dimension lying in the direction of the length of the boiler. The shell plates are 3/8 in. thickness, and the reenforcing ring is to be 1/2 in. thick, single riveted, driven size of rivets 7/8 in. What is the required width of ring?

Using formula (1) the statement becomes:

$$W = \frac{11 \times .375}{2 \times .5} + .875 = 5.0 \text{ in., } \textit{answer.}$$

A manhole opening is 11×15 in.; the 11-in. dimension lying in the direction of the length of the boiler. The shell plates are 1/2 in. and two 1/2-in. rings are to be used, double riveted, driven size of rivets 7/8 in. What width of rings is required?

In this case formula (4) is to be used. The statement becomes:

$$W = \frac{11 \times .5}{4 \times .5} + 2 \times .875 = 4.50 \text{ in., } \textit{answer.}$$

How many 7/8-in. driven rivets are to be used in a single ring 1/2 in. thick, 5 in. wide, 60,000 lb. tensile

strength, 38,000 lb. shearing stress of the rivets, ring to be single riveted.

Formula (5) is to be used, and statement becomes: The net section of the ring is $(5-.875)\times.5=2.0625$ sq. in.

$$N=\frac{4\times60000\times2.0625}{38000\times(.875^2\times.7854)}=21 \text{ required, say } 20.$$

When the total number of rivets is an odd number, change it so that there will be an equal number of rivets on each side of the center. A number divided by 4 may be used.

How many 1-in. driven rivets should be used in a manhole reenforced by two rings 3/4 in. thick and 4 1/2 in. wide, single riveted, 38,000 lb. shearing stress per square inch section, and 60,000 lb. tensile strength of the material of which the rings are made.

Formula (6) is to be used. The net section of the ring is $(4.5-1)\times.75=2.625$ sq. in.

$$N=\frac{8\times60000\times2.625}{1.85\times38000\times(1^2\times.7854)}=22.8, \text{ say } 24 \text{ rivets.}$$

(1.85 is a constant for rivets in double shear.)

Rings whose width is found by the formulas in this group will have a *total net cross-sectional area* equal to that of the metal taken from the shell to make the opening. The *total* net section means for a *single* ring, *twice* the net sectional area of the ring, and for double rings, *four* times the net sectional area of one ring.

The formulas for finding the number of rivets to be used make the resistance of the rivets to shearing equal to the resistance to tensile stress of the metal in the rings.

The constant 4 that appears in formula (5) is obtained from two sectional areas for each half of the ring, and all the rivets in each half. Therefore in both halves of the ring, and when considering the total number of rivets in the whole ring, there are four sections to consider.

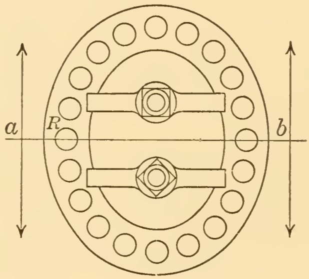


FIG. 13.—Reenforcing ring. Diagram illustrating direction of stress in reenforcing rings, and the origin of the constants 4 and 8 which appear in the formulas. Arrows *a* and *b* show direction of stress borne by each half of the ring, and the rivets in each half.

With two rings, there are four sectional areas for each half of the two rings, or twice as much as with one ring, therefore the constant becomes 8 in formula (6).

HEATING SURFACE OF BOILERS

The heating surface of a steam boiler is that part exposed to the action of the hot gases from the furnace. Finding the heating surface is a matter of mensuration, and it is expressed in square feet.

To find the heating surface of a return-tubular boiler, multiply two-thirds of the circumference of the shell by

the length, both in inches; multiply the number of tubes by the circumference of one tube and by its length, also in inches; take two-thirds of the area of each tube sheet minus the area due to the tube openings in square inches, add the products together and divide the sum by 144 to convert into square feet.

To find the heating surface of a vertical tubular boiler, multiply the circumference of the fire box by its height above the grate, both in inches; multiply the number of tubes by the circumference of one and by its length, also in inches; find the area of the lower tube sheet minus the area of tube openings, all in square inches; add the results together, and divide the sum by 144 to convert into square feet.

To find the heating surface of water-tube boilers, multiply the number of tubes by the circumference of one and by its length, both in inches; find the exposed area in square inches of one set of headers; find the number of square inches in one-half of the steam drum or drums as the case may be; add the values together, and divide by 144 to convert into square feet.

The true heating surface of a tube is the side exposed to the hot gases; the inner surface in a fire tube, and the outer surface in a water tube.

The following example illustrates the rule for finding the heating surface of a steam boiler.

Example.—What is the total heating surface of a horizontal return tubular boiler 60 in. diameter and 12 ft. long, with 80 tubes 2 in. diameter?

For the sake of simplicity consider the 2-in. dimension as the *inside* diameter of the tubes.

The statement will be:

Circumference of the shell $= 60 \times 3.1416 = 188.496$ in.

Length of shell.............................. $12 \times 12 = 144$ in.

Heating surface of the shell...... $188.496 \times 144 \times 2/3 = 18095.616$ sq. in.

Circumference of one tube.............. $2 \times 3.1416 = 6.2832$ in.

Heating surface of all the tubes..................... $80 \times 144 \times 6.2832 = 72382.46$ sq. in.

Area, in square inches, of one head........ $60^2 \times .7854 = 2827.44$.

Two-thirds area of both heads.............. $2/3 \times 2 \times 2827.44 = 3769.92$.

Area, in square inches, through all the tubes....... $2^2 \times .7854 \times 80 = 251.328$.

Total heating surface $\dfrac{18095.616 + 72345.6 + 3769.92 - 2 \times 251.328}{144}$

$= \dfrac{93708.48}{144} = 650.75$ sq. ft.

Grate Area

The required grate surface in any given boiler depends upon the rate of combustion of the fuel, the quantity of water evaporated per pound of the fuel used, and the total weight of steam generated per hour.

To find the grate area required for any given plant:

$A = \dfrac{W}{r \times w}$ in which the values are:

A = area of grate surface in square feet.

W = weight of steam required per hour.

w = pounds of water evaporated per pound of fuel consumed.

r = rate of combustion in pounds per square foot of grate per hour.

Example.—A battery of boilers is required to generate 7000 lb. of steam per hour, consuming 15 lb. of coal on each square foot of

grate per hour. Assume the evaporation to be 7 lb. of water per pound of the coal used, what grate area will be required?

Applying the formula, the statement becomes:

$$A=\frac{7000}{15\times 7}=66.66 \text{ sq. ft.}$$

As the average evaporation per pound of fuel is different for different boilers, the formula given is approximate only, but for practical work it is considered suitable to use.

Corrugated Furnaces

To find the safe working pressure of steel corrugated flues and furnaces:

$$P=\frac{C\times T}{D}$$

In which:

$P=$ safe pressure in pounds per square inch.

$T=$ thickness of metal of which the furnace is made.

$D=$ mean diameter of furnace in inches.

$C=$ a constant, as follows:

15,600 for Morrison flues, under United States Rules.

14,000 for Morrison flues under British and Canadian Rules, and also for Purvis, Fox, and Brown flues, under United States, British, and Canadian Rules.

The mean diameter of a Morrison flue, under United States Rules, is the least inside diameter of flue plus

2 in. Under British and Canadian Rules, the mean diameter is the diameter at the bottom of the corrugations, as measured from the outside. The Fox type is measured in the same way for the mean diameter by the United States, British, and Canadian Rules.

Example.—What is the safe working pressure on a Morrison furnace flue 36 in. mean diameter, with a thickness of metal of 3/8 in. The value of C in the question is to be taken as 15,600.

Applying the formula:

$$P = \frac{15600 \times .375}{36} = 162.5 \text{ lb.}$$

For furnaces other than corrugated, different authorities give different formulas, according to size and design. For plain flues made in sections not more than 8 ft. in length, and with the ends of each section flanged and riveted, with a ring between the flanges, this rule may be used:

$$P = \frac{89600 \times T^2}{L \times D}$$

In which:

$P =$ safe working pressure in pounds per square inch.
$T =$ thickness of flue in inches.
$D =$ diameter of flue in inches (outside diameter).
$L =$ length of section in *feet.*

Example.—A plain furnace flue is 24 in. diameter and 3/8 in. thick, and made in sections of 6 ft. long. What pressure is safe to carry?

Applying the formula:

$$P = \frac{89600 \times .375^2}{6 \times 20} = 105.0 \text{ lb.}$$

The collapsing pressure of steel tubes of sizes from 3 to 10 in. may be found from the following formulas.

$$P=86{,}670\frac{T}{D}-1386 \qquad (1)$$

And

$$P=1000\left(1-\sqrt{1-1600\frac{T^2}{D^2}}\right) \qquad (2)$$

The first one is to be used when the value of P is greater than 581 lb. The second one is to be used when the value of P is less than 581 lb.

The values of the letters in both formulas are:

$P=$ collapsing pressure in pounds per square inch.
$T=$ thickness of tube in inches.
$D=$ outside diameter of tube in inches.

These formulas are based on experiments conducted at the National Tube Works, McKeesport, Pa.

The factors of safety that may be used for tubes varies from 4 to 7, according to surrounding conditions. For instance, in a case where considerable damage to life and property might result from the collapsing of a tube, a factor of safety of 7 should be used.

Example.—What is the presumed collapsing pressure of a 3.5-in. lap-welded steel tube of .12-in. thickness? What pressure may be safely carried where a moderate amount of loss would occur from the failure of a tube?

First try formula (1).

$$P=86{,}670\times\frac{.12}{3.5}-1386$$

$$=(86{,}670\times .0343)-1386=1586.781 \text{ lb. per square inch.}$$

Formula (1) proves to be the correct one to use, as the result is greater than 581 lb. specified.

The safe pressure to carry, using 5 as a factor of safety under the conditions stated, will be:

$$\frac{1586.781}{5}=317+ \text{ lb. per square inch.}$$

Under United States Rules, lap-welded boiler tubes from 1 in. to 6 in., inclusive, may be of any length, and may be allowed an external safe working pressure up to and including 225 lb. per square inch. This gives a liberal factor of safety.

Horse-power of Boilers

In reality there is no such thing as horse-power of steam boilers, but the term has come into use and therefore requires definition. In order to have a definite value by which to compare boiler performances under different conditions, the American Society of Mechanical Engineers decided that a standard boiler horse-power should be equal to the absorption of 33,330 B.t.u. by the water in the boiler. This is based on the evaporation of 34 1/2 lb. of water per hour from a temperature of 212° F. into steam of the same temperature and corresponding pressure, that of the atmosphere.

As the latent heat of steam at atmospheric pressure is required to convert 1 lb. of water at 212° F. into steam of 212° F., then the latent heat in B.t.u.'s times the number of pounds of water chosen as the standard, gives the value, thus:

$$966.1\times 34.5=33,330 \text{ B.t.u.}$$

The standard boiler horse-power is found by the following formula:

$$H.P. = \frac{W \times (H - t + 32)}{33330}$$

In which, $H.P.$ = the horse-power.

W = weight of water in pounds actually evaporated per hour.

H = total heat of steam above 32°, at the pressure of evaporation.

t = the temperature of the feed water.

Example.—A boiler evaporates 3000 lb. of water per hour from a feed-water temperature of 100° F. into steam of 85 lb. gage pressure, what is the standard horse-power?

The absolute steam pressure is $85 + 15 = 100$ lb. in round numbers. Referring to a table of properties of saturated steam, it will be found that the total heat of steam at 100 lb. absolute pressure is, in round numbers, 1182 B.t.u. The statement becomes:

$$H.P. = \frac{3000 \times (1182 - 100 + 32)}{33330} = 100+\text{horse-power.}$$

Even values have been taken instead of exact ones, in order to simplify the operation. It must also be remembered that different steam tables in use will give slightly different values, but for purely practical work, the difference need not be considered.

Horse-power rating of boilers is sometimes based upon the number of square feet of heating surface, varying from 6 to 20 square feet per horse-power, according as to the type of boiler as follows:

Water-tube boilers, 10 to 12 sq. ft. per horse-power.

Return tubular boilers, 12 to 15 sq. ft. per horse-power.

Vertical tubular boilers, 15 to 20 sq. ft. per horse-power.

Flue boilers, 8 to 12 sq. ft. per horse-power.

Plain cylinders, 6 to 10 sq. ft. per horse-power.

These values are arbitrary and serve only as a guide in buying and selling. Different manufacturers had in the past adopted different values, there being no fixed standard until recently, when the Boiler Manufacturers' Association adopted 10 sq. ft. of heating surface to a horse-power, in horizontal tubular boilers.

Ratio of Heating Surface to Grate Area

The ratio between the heating surface and grate surface varies with the type of boiler and also with the rate of combustion.

$$\text{Ratio} = \frac{\text{heating surface}}{\text{grate surface}}$$

$$\text{Ratio} = \begin{cases} \text{Water-tube boilers, 35 to 40.} \\ \text{Horizontal tubular, 25 to 35.} \\ \text{Vertical boilers, 25 to 30.} \\ \text{Locomotive boilers, 50 to 100.} \\ \text{Flue boilers, 25 to 35.} \\ \text{Plain cylinder boilers, 12 to 15.} \end{cases}$$

Equivalent Evaporation

Equivalent evaporation from and at 212° F. means the quantity of water that would be converted into steam of 212° F. and at atmospheric pressure, from a feed-water temperature of 212° F. as compared with the actual evaporation under certain conditions, for any given case.

Equivalent evaporation reduces actual evaporation to a standard basis, from which comparisons can be made in boiler trials.

The formula is as follows:

$$W = \frac{w \times (H - t + 32)}{970.4}$$

In which the letters have the following values:

W = equivalent evaporation from and at 212° F. in pounds.

w = actual evaporation in pounds.

H = the total heat of steam above 32° F. at the pressure of evaporation, as found in the steam tables.

t = the temperature at which the feed water enters the boiler; 970.4 latent heat of steam at atmospheric pressure according to Marks and Davis's tables.

Example, illustrating the application of the formula.

A certain boiler generates 3000 lb. of dry steam per hour at a pressure of 150 lb. gage from a feed-water temperature of 200° F. What quantity of water would have been evaporated had the feed water been delivered at 212° F. and converted into steam of 212° F. and at atmospheric pressure?

According to Marks and Davis's steam tables, the total heat of steam at 150 lb. gage pressure, or 165 lb. (in round numbers) absolute pressure is 1195.0 B.t.u. Applying the factors, the statement becomes:

$$W = \frac{3000 \times (1195.0 - 200 + 32)}{970.4} = 3175 \text{ lb., closely.}$$

If steam tables be used other than Marks and Davis's, slightly different results will be obtained. Before these tables were devised, the constant used was 965.7 and

sometimes 966.1 instead of 970.4. These values represent the latent heat of steam at atmospheric pressure, according to the tables from which they are derived.

It may be stated that, excepting in cases where extreme accuracy is required, it makes little difference to the engineer as to which value he uses, nor from which steam table he procures any of the values used in steam calculations. As between the three latent heat constants given, a difference in results obtained will be less than 1 per cent., which is sufficiently close for all calculations in the realm of the practical operating engineer.

In the formula, the quantity $\frac{(H-t+32)}{970.4}$ that changes the actual evaporation of 1 lb. of water to equivalent evaporation from and at 212° F. is known as *the factor of evaporation.* In the example given, the factor of evaporation is:

$$\frac{1195-200+32}{970.4}=1.0583$$

and when multiplying the actual total evaporation by the factor, the equivalent evaporation is obtained thus: $3000 \times 1.0583 = 3174.90$ which, as before given, would be called 3175 lb. in round numbers.

The factor for any other values of steam pressure and feed-water temperature will be obtained and used in the same way as just illustrated.

Boiler Efficiency

The efficiency of a boiler plant is the ratio of the difference between the heat in the steam (delivered by the boiler) and the heat in the feed water, to the heat that

would be developed by the perfect combustion of the fuel.

The following example illustrates the foregoing definition:

The heat of combustion of a certain fuel is known to be 14,000 B.t.u. pe· pound; the number of pounds of such fuel used at a boiler test was 3500 per hour; the evaporation was 30,000 lb. of feed water per hour, into steam of 90 lb. gage pressure, or 105 lb. absolute; the temperature of the feed water was 80° F. It is required to find the efficiency of the boiler.

The method of operation is as follows:

The total heat in 1 lb. of steam at 105 lb. absolute is 1187.2 B.t.u. (from Marks and Davis's tables).

Temperature of feed water $=80°$ F., and $80-32=48$ assumed B.t.u. contained in the feed water above 32° F. Then: $1187.2-48=1139.2$ B.t.u. and $1139.2\times 30{,}000=$ 34,176,000 B.t.u. absorbed by the water.

As each pound of coal contains 14,000 B.t.u. and as the total coal consumed was 3500 lb. the total heat supplied $=14{,}000\times 3500=49{,}000{,}000$ B.t.u.

The efficiency is:

$$\frac{34176000\times 100}{49000000}=69.74 \text{ per cent.}$$

A Short Method to Find the Commercial Efficiency of Boiler and Furnace Combined

Expressed in terms of cost of evaporating 1000 lb. of water from and at 212° F.

$$c=\frac{C}{2\times e}$$

In which the values are as follows:

c = cost of evaporating 1000 lb. of water from and at 212° F.

C = cost of coal per ton of 2000 lb.

e = the evaporation per pound of coal, from and at 212° F.

2 = a constant, to reduce the values to a common basis of 1000.

Example, illustrating the formula:

A certain coal costs $2.00 per ton of 2000 lb., and is known to actually evaporate 8 lb. of water per pound of coal consumed on the grates. The factor of evaporation—from the method previously explained—is found to be 1.0583. Therefore the equivalent evaporation from and at 212° F. is $1.0583 \times 8 = 8.4664$ lb. of water per pound of coal.

Then, $c = \frac{2.00}{2 \times 8.4664} = \$.118$; or 11 8/10 cents per lb.

As a check on the accuracy of the foregoing proceed thus:

Coal at $2.00 per 2000 lb., costs per pound, $\frac{200}{2000} = .10$ cents.

Each 8.4646 lb. of water evaporated from and at 212° F. costs .1 or 1/10 of a cent. As the cost for evaporating 1000 lb. is required, then

$$\frac{1000}{8.4646} = 118.1 \text{ lb. of coal required}$$

and $118.1 \times .1 = 11.81$ cents, as found before.

How to Analyze a Boiler Trial Report

The following is the report of a boiler trial which was held to determine the efficiency under given conditions.

TRIAL OF A 1000 H.P. BOILER

	1.	Kind of trial	running start and stop.
	2.	Duration of trial	24 hours.
Surfaces.	3.	Grate surface	113.6 sq. ft.
	4.	Total heating surface	10,000 sq. ft.
	5.	Ratio of heating surface to grate surface	88.0
Pressures.	6.	Average pressure per square inch, gage	81.0 lb.
	7.	Average atmospheric pressure per sq. in.	14.84 lb.
	8.	Average absolute pressure per sq. in.,	95.84 lb.
	9.	Force of draft (column of water)	.21 in.
Temperatures.	10.	Temperature of the external air	30° F.
	11.	Temperature of the fire room	93° F.
	12.	Temperature of the feed water before entering the boiler	37.3° F.
	13.	Temperature of escaping gases after leaving the boiler	407° F.
	14.	Temperature of the steam	324+° F.
	15.	Moist coal consumed	76,687 lb.
	16.	Moisture in the coal	4.12 per cent.
	17.	Dry coal consumed	73,528 lb.
	18.	Total dry refuse	8069 lb.
	19.	Total dry refuse	10.97 per cent.
	20.	Total combustible	65,459 lb.
	21.	Dry coal consumed per hour	3064 lb.
	22.	Combustible consumed per hour	2727 lb.
	23.	Percentage of moisture in the steam	0.6013 per cent.
	24.	Number of B.t.u. in 1 lb. dry coal	14,500.
	25.	Number of B.t.u. in 1 lb. combustible	15,425.
B.t.u.	26.	Heat absorbed by the boiler, per pound of steam generated	1175.6 B.t.u.
	27.	Total B.t.u. absorbed by the boiler	774,880,282.
	28.	Heat units imparted to the boiler per pound of dry coal	10,538.
	29.	Heat units per pound of combustible	11,838.

Efficiency.	30. Efficiency of the boiler, based upon dry coal (approximately)	72.7 per cent.
	31. Efficiency of the boiler, based upon combustible	76.77 per cent.
	32. Factor of evaporation	1.22.
Water	33. Total water fed to boiler	663,124 lb.
	34. Water actually evaporated, corrected for quality of steam	659,136 lb.
	35. Equivalent water from and at 212° F. boiler only	804,146 lb.
	36. Equivalent water from and at 212° F. per hour, boiler only	33,506 lb.
	37. Water actually evaporated per pound of dry coal	8.96 lb.
	38. Water evaporated per pound of combustible	10.07 lb.
Horse-power.	39. Horse-power, basis 34 1/2 lb. from and at 212° F.	971.
	40. Number of square feet heating surface per horse-power	10.3.
	41. Horse-power per square foot of grate	8.53.
	42. Builder's rating, horse-power	1000.

In the following, some of the results are not exact as far as actual numerals are concerned; this is due to disregarding too small decimal values, and using round numbers instead as far as consistent. For practical purposes the values are sufficiently close; the object in view is to explain and illustrate *how* the values are found, rather than to give an exhibition of exact arithmetical operations.

The first two items are self-explanatory.

The grate surface (item 3) is found by multiplying together the length and width in feet, which gives the area in square feet. The heating surface (item 4) is the sum total of all surfaces which are in contact with the

hot gases on one side and water on the other side. The method of finding the heating surface has been explained in a previous section.

The ratio of heating to grate surface (item 5) is found by division thus: $10{,}000 \div 113.6 = 88.0$. That is, item 4 is to be divided by item 3.

The pressure of the atmosphere is noted (item 7) as 14.84 lb. per square inch. The pressure of the atmosphere is usually considered as 14.7 lb. per square inch, and in ordinary calculations as 15 lb. in round numbers, at sea level. As a matter of fact the pressure of the atmosphere is constantly changing as indicated by the barometer. The way 14.84 was found is this: the height of the barometer divided by 2.04 equals atmospheric pressure. The 2.04 factor is the height of mercury in inches that is equal to 1 lb. per square inch.

As an example, the barometer indicates 29.5 in., and $29.5 \div 2.04 = 14.4$ lb. per square inch. Applying this, $14.84 \times 2.04 = 30.2+$ in., which was the barometer reading at the time of the trial.

Absolute pressure is measured from a perfect vacuum, and is gage pressure plus atmospheric pressure, in this instance $81 + 14.84 = 95.84$ lb., item 8.

The force of draft in column of water is 0.21 in. A draft gage is used for the purpose, one (1) ounce pressure per square inch is equal to 1.73 in. of water as registered by the gage. The value 1.73 is found like this:

34 ft. = 408 in.
408 in. = 14.7 lb. (at sea level).
14.7 lb. = 235.2 oz.

Therefore, 1 oz. pressure $= \frac{1}{235.2}$ of 408 in., or

$408 \div 235.2 = 1.73$ in. of water as indicated by gage. As a matter of convenience, the atmospheric pressure at the time of the trial is considered as having been 14.7 lb. instead of 14.84, and the draft pressure in ounces is $0.21 \div 1.73 = 0.121$ or $\frac{121}{1000}$ of an ounce.

The temperature of the steam (item 14) is found from a steam table, of which there are several in use; the one in particular gave the value of 324° F. corresponding to the pressure.

The number of pounds of moist coal consumed was 76,687 as actually weighed. As the coal contained 4.12 per cent. of moisture, 4.12 per cent. of the 76,687 must be subtracted, which gives as a remainder 73,528 lb. of dry coal consumed (item 17), thus: $76{,}687 \times 4.12 = 3159.50$ and $76{,}687 - 3159.50 = 73{,}527.5$, say 73,528 lb. in round numbers.

The moisture per cent. in the coal (item 16) is found like this: Exactly 100 lb. of coal, from the pile to be used during the trial, is placed in a bag or a box, and subjected to a good heat, such as would obtain on top of the boiler in operation, until it is thoroughly dried out. The coal is again weighed, and in the case under consideration was found to be 95.88 lb.; then, $100 - 95.88 = 4.12$ lb. moisture was evaporated, and this is 4.12 per cent. of the original weight.

The total dry refuse—the non-combustible part of the coal—was 8069 (item 18) which is: (item 19) $8069 \div 73{,}528 = 10.97$ per cent. The total combustible (item 20) was: 73,528 lb. dry coal minus 8069 dry refuse = 65,459 lb. combustible.

The total quantity of dry coal consumed was 73,528; for one hour the quantity was $73{,}528 \div 24 = 3064$ lb. (item 21).

The combustible for one hour is found in the same manner.

The percentage of moisture in the steam (item 23) 0.6013 was determined by the use of a calorimeter.

The heat value of the coal was determined by an analysis and test in a laboratory equipped with apparatus for that and similar purposes. In item 24 it is given as 14,500 B.t.u. per pound dry coal, and in item 25, 15,425 B.t.u. per pound of combustible.

In item 26 the heat absorbed by the boiler per pound of steam generated was found like this:

By referring to a steam table, the latent heat of steam at 95 lb. absolute pressure—which the report records—is found to be 886.7 B.t.u., while the sensible heat is 323.89 and $886.7 + 323.89 = 1210.59$, from which is to be subtracted the given temperature of the feed water, 37.3° F., which gives $1210.59 - 37.3 = 1173.29$ B.t.u. This method is approximate only. The following is more nearly accurate.

The total heat of steam above 32° F. at 95 lb. absolute pressure is 1180.7, as found in the steam table used in connection with this particular test. The temperature of the feed water above 32° F. was $37.3 - 32 = 5.3$, and $1180.7 - 5.3 = 1175.4$ B.t.u., which more nearly agrees with the item in the report.

The difference of 2.11 which exists between the two results is accounted for in this way:

The temperature of steam at 95 lb. absolute pressure

is 323.89° as found from the table used at that time. But strictly speaking, calculations involving the steam tables are based on water from 32° F. and not from zero on the Fahrenheit scale, as is sometimes done. Ignoring this will give approximate results only. Then, 323.89 $-32=291.89$; but the heat of the liquid as found in the table is 294.0 and $294.0-291.89=2.11$, the difference before referred to. This difference is due to degrees temperature and B.t.u.'s not being exactly the same in value. However, the difference in the range of the steam tables is so small, that for all practical purposes where extreme accuracy is not demanded, degrees temperature may be used instead of heat in the liquid values. It is only as a matter of convenience that degrees temperature and units of heat are sometimes considered synonymous.

The total heat units absorbed by the boiler during the trial was 774,880,282 (item 27) found like this: As 1175.6 is the heat units per pound of steam generated by the actual quantity of water evaporated (corrected for the quality of the steam) which appears as 659,136, then $1175.6 \times 659{,}136 = 774{,}880{,}282$ total B.t.u. absorbed.

The heat units imparted to the boiler per pound of *dry coal* are found to be 10,538, found like this:

$$\underset{\text{Total B.t.u.}}{774{,}880{,}282} \div \underset{\substack{\text{Dry coal}\\ \text{consumed}}}{73{,}528} = 10{,}538 \text{ B.t.u.}$$

per pound of *dry coal* (item 28).

The heat units imparted per pound of combustible

is found in a similar manner, only using the pound combustible as a divisor thus:

$$774{,}880{,}282 \div 65{,}459 = 11{,}838 \text{ B.t.u.}$$

per pound of combustible (item 29).

The efficiency of the boiler based upon dry coal (item 30) is found by dividing the heat units per pound of dry coal by the theoretical heating value, which is taken as 14,500 B.t.u. thus: $10{,}538 \div 14{,}500 = .7267$ which is 72.67 per cent.

The efficiency based upon the combustible is:

$$\underset{\text{B.t.u. per lb. combustible}}{11{,}838} \div \underset{\text{Theoretical value}}{15{,}425} = .7675$$

which is 76.75 per cent. (item 31); the efficiency of the boiler is the ratio of the heat utilized to that supplied.

The factor of evaporation (item 32), 1.22, was found like this:

$$\frac{1181 - 5.3}{965.7} = 1.22$$

in which 1181 = B.t.u. in steam at 95 lb. pressure absolute.

$$965.7 = \text{B.t.u.}$$

required to evaporate 1 lb. of water from and at 212° F.

$$5.3 = (37.3 - 32)$$

The factor of evaporation means, that for every pound of water actually evaporated under the prevailing conditions at the time of the trial, 1.22 or 1 22/100 lb. of water would have been evaporated had the feed-water

temperature been 212° F. and had the pressure been that of the atmosphere.

Item 33 gives the total number of pounds of water fed to the boiler during the trial, as found by actually weighing it.

Item 34 is found by multiplying the total weight of water fed to the boiler by the percentage of moisture as found in the steam expressed decimally, and then subtracting that value from the original quantity, thus:

$$663{,}124 \times .006013 = 3988- \text{ lb.}$$

moisture in the steam during the whole test, and $663{,}124 - 3988 = 659{,}136$ lb. actually evaporated.

The equivalent evaporation from and at 212° F. (item 35) is found thus: actual evaporation times factor of evaporation, or

$$659{,}136 \times 1.22 = 804{,}146 \text{ lb.}$$

The equivalent evaporation per hour (item 36) is found thus:

$$804{,}146 \div 24 = 33{,}506 \text{ lb.}$$

The quantity of water actually evaporated per pound of dry coal consumed is 8.96 lb. (item 37) as found from dividing the total water actually evaporated by the total dry coal consumed. Or,

$$659{,}136 \div 73{,}528 = 8.96 \text{ lb.}$$

The water actually evaporated per pound of combustible (item 38) is found in a similar way, thus:

$$659{,}136 \div 65{,}459 = 10.07 \text{ lb.}$$

Take the 8.96 lb. water per pound of dry coal, and multiply it by the factor of evaporation 1.22 thus: $8.96 \times 1.22 = 10.94$ lb. of water from and at 212° F. on the *dry coal* basis.

So, also in relation to the combustible, $10.07 \times 1.22 = 12.29$ lb. water from and at 212° F. on the *combustible* basis.

The horse-power is found (item 39) by dividing the pounds of water evaporated per hour from and at 212° F. by the standard 34.5 thus: $33{,}506 \div 34.5 = 971$ H.P. The builder's rating was given as 1000 H.P.

This is in excess of the actual power developed at the trial by $1000 - 971 = 29$ H.P. and this expressed as a percentage is 2.9 found thus:

$$29 \div 1000 = .029 \text{ or } 2.9 \text{ per cent.}$$

By dividing the 10,000 sq. ft. heating surface by 971, the quotient obtained is 10.3 (nearly) which is the number of square feet per horse-power (item 40).

And 113.6 sq. ft. grate surface divided into 971 gives 8.53 H.P. per square foot of grate (item 41).

PART II

MISCELLANEOUS APPLICATIONS OF BOILER ARITHMETIC

MISCELLANEOUS APPLICATIONS

Bursting Pressure of Pipe

$$P=\frac{2t\times S}{D}$$

In which, P = bursting pressure, pounds per square inch.
t = thickness in inches.
S = tensile strength of the metal in pounds per square inch.
D = internal diameter of pipe in inches.

Example.—Find the bursting pressure of a 10-in pipe, 0.366 in. thick, actual internal diameter 10.019 in. tensile strength of the metal taken as 50,000 lb.

$$\frac{2\times .366\times 50{,}000}{10.019}=3653 \text{ lb.}$$

For the safe working pressure, a factor of safety of not less than 10 should be used and preferably more. Therefore $\frac{3653}{10}=365.3$ lb. safe working pressure.

The bursting pressure being given, to find the thickness of metal the formula is transposed thus:

$$t=\frac{D\times P}{2\times S}$$

Using the terms of the same question the statement becomes:

$$t=\frac{10.019\times 3653}{2\times 50000}=0.366 \text{ in.}$$

When the exact tensile strength is not known, assume 50,000 lb. for steel, and 40,000 lb. for iron pipe.

The actual bursting pressure of pipes, as found from tests, is less than that found from the foregoing formula, and this makes it necessary that a liberal factor of safety be used.

The Force Tending to Tear Asunder

The force tending to tear asunder two cylinders of

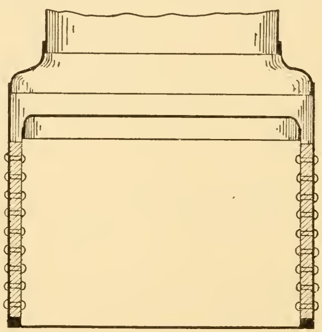


Fig. 14.—Lower part of Manning boiler. Diagram showing area against which the pressure acts, tending to tear apart the two cylinders.

different diameters as illustrated in Fig. 14, which represents a portion of the Manning boiler, the pressure being applied in the annular space between the two,

depends upon the *cross-sectional area of the space*, and is *independent of the diameter.* The force tending to break the plates is the area of the sheets in the cross-hatched portion multiplied by the pressure per square inch. This force is resisted by the four plates, the outer and inner plates being in tension.

The foregoing is based on the condition that the two cylinders are stayed to each other as shown in the illustration.

Method of Finding the Fuel Cost of Evaporating 1000 Lb. of Water

DATA

Assume that in a given case, 21 tons of coal were consumed in a certain time, and that the cost of the coal was $3.00 per ton of 2000 lb.

Also assume that in the same period of time, 398,616 lb. of water were evaporated into steam at a pressure of 100 lb. per square inch absolute, the temperature of the feed water being 200° F.

The number of heat units in a pound of water at 200° F. is 168.713 (as found from steam tables).

The number of heat units in 1 lb. of steam at 100 lb. pressure per square inch absolute is 1181.866 (as found in tables).

The number of heat units required to convert 1 lb. of water into steam at atmospheric pressure, 14.7 lb. per square inch, is 966 (in round numbers), and this is the *latent* heat of steam at 14.7 lb. pressure per square inch.

METHOD OF OPERATION

First, find the factor of equivalent evaporation to reduce the problem to the standard basis of, from and at 212° F. Employing the given values:

$$\frac{1181.866-168.7}{966}=1.048 \text{ factcr of evaporation.}$$

The statement of the problem becomes:

$$\frac{\overset{\text{Tons}}{21} \times \overset{\text{Cents}}{300} \times \overset{\text{Lb. water}}{1000}}{\underset{\text{lb. water.}}{398616} \times \underset{\substack{\text{Factor of}\\ \text{evap.}}}{1.048}} = 14.6 \text{ cts.}$$

or, $0.146, cost of evaporating 1000 lb. water under the assumed conditions. The actual cost will be more than that found above, for labor, depreciation, cost of water, taxes, insurance, and interest on investment of the plant as a whole, enters the calculation where precision is required and over all charges are to be made.

Safe Working Pressure for Cylindrical Cast-iron Vessels with Flat Cast-iron Heads

RULES PRESCRIBED BY THE UNITED STATES INSPECTORS OF STEAM BOILERS

When evaporators, feed-water heaters and separators are made of good cast iron, the shells cylindrical and the ends flat, the castings sound and of uniform thickness, the working pressure shall not exceed that found by the following formulas: For finding the safe pressure on the *flat* surface this is the formula to be used:

$$P=\frac{20000 \times T^2}{D^2}$$

For the cylindrical part of the vessel this is the formula to be used:

$$P = \frac{3500(T - 1/4)}{D}$$

And to find the thickness of metal required, having the other values given, use these formulas:

For the flat heads

$$T = \sqrt{\frac{P \times D^2}{20000}}$$

and for the cylindrical shell

$$T = \frac{P \times D}{3500} + 1/4.$$

In the formulas given the value of the letters stand like this:

P = safe working pressure in pounds per square inch.

T = thickness of metal in inches, *provided* the thickness of the *ends or heads* of such vessels shall not be less than 3/8 in.

D = *inside* diameter of the vessel in inches. When the *ends or heads* are bolted to the shell then D = the diameter of the bolt circle. When the pressure is to be determined for a part of a flat surface which is square or rectangular, the value of D in the flat surface formula shall be the *diagonal* of the square or rectangle. The numbers 20,000 and 3500 are constants, evidently empirical, and found from experiment.

Equivalent Boiler Performance

A boiler is sold on a guarantee that it will evaporate 12 lb. of water per pound of combustible, from and at 212° F., with coal having a heat value of 14,500 B.t.u. per pound of coal, and 8 per cent. ash.

When the test was made the average feed-water temperature was 180° F., average steam pressure 70 lb., heat value of the coal used 12,000 B.t.u. per pound, and the ash content 8 per cent.; water evaporated, 9 lb. per pound of coal consumed.

It is required to ascertain how the test compares with the guarantee.

Under the conditions of the guarantee the coal is .08 ash, and $1-.08=.92$ combustible.

If 1 lb. of coal contains 14,500 B.t.u., 1 lb. of combustible will contain $\frac{14500}{.92}=15{,}760$ B.t.u.

With this it is agreed to evaporate 12 lb. of water from and at 212° F. or 1 lb. under those conditions for each $\frac{15760}{12}=1313$ B.t.u. supplied.

The temperature corresponding to 85 lb. absolute (70 lb. gage in round numbers) is 316.3° F.

To raise 1 lb. of water from 32° to 316.3° requires 286.30 B.t.u.

To raise 1 lb. of water from 32° to 180° requires 147.88 B.t.u.

To raise 1 lb. of water from 180° to 316.3° requires 138.42 B.t.u.

To evaporate 1 lb. of water from and at 316.3° requires 897.10 B.t.u.

To raise 1 lb. of water from 180° to 316.3°, and evaporate it at 85 lb. pressure absolute, requires 1035.52 B.t.u.

The coal used at the test contained 12,000 B.t.u. per pound, and as there was 8 per cent. of ash, $\frac{12000}{.92}=$ 13,043 B.t.u. per lb. of combustible.

Under the conditions of the test, 1035.52 B.t.u. were required per pound of water evaporated into steam. Then, $\frac{13043}{1035.52}=12.59$ lb. water evaporated per pound of combustible, as compared with the guarantee of 12 lb. of water per pound of combustible.

Efficiency of Diagonal Seam

Assume a seam such as that shown in Fig. 15; the shell

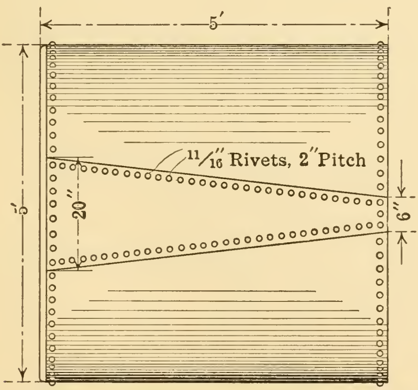


Fig. 15.—Diagonal seam.

is 60 in. diameter, made of 5/16-in. plate having a tensile strength of 60,000 lb. per square inch. The seam is 20

in. wide at one end and 6 in. at the other. The rivet holes are 11/16 in. in diameter and the pitch is 2 in.; the rivets are of steel. Assuming the efficiency of the joint to be 41 per cent., what increase in efficiency is obtained by the seam being slightly diagonal to the axis of the shell?

To find this in a simple manner, multiply the length of the seam in inches by 41, the efficiency of the joint, and divide the product by 60, which is the length in inches measured along a line parallel to the axis of the cylinder. Thus: As the seam, measured along the rivet center of the diagonal seam is 60.4 in., then,

$$\frac{60.4 \times 41}{60} = 41.27 \text{ per cent.}$$

efficiency of the diagonal seam, a gain of .27 per cent.

Collapsing Strength of Cone-shaped Flue

A cone-shaped flue has a greatest diameter of 36 in. and a least diameter of 12 in. and a length of 20 in. It is made of 5/16-in. steel plate of 60,000 lb. tensile strength per square inch. It is required to find what collapsing pressure such a flue will safely withstand. The construction is shown in Fig. 16.

It is customary in short cones to take the mean diameter, in this case,

$$\frac{36+12}{2} = 24 \text{ in.,}$$

and calculate the collapsing strength as follows:

Hutton's rule is,

$$\frac{T^2 \times C}{D \times \sqrt{L}} = P$$

The values of the letters are:

T = Thickness of plate in thirty-seconds of an inch

D = External diameter of the shell in inches.

L = Length of shell in inches.

C = 660 for mild steel plates.

P = Collapsing pressure.

Applying the formula, the statement becomes:

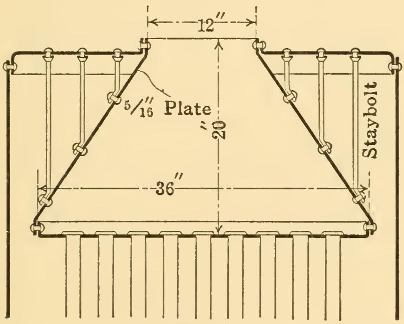


FIG. 16.—Cone-shaped flue.

$$P = \frac{10 \times 10 \times 660}{24 \times \sqrt{20}} = 614.9 \text{ lb.}$$

The collapsing pressure must be divided by the factor of safety, and in view of high temperatures and wear and tear, this should be 6; the allowable pressure will be, $\frac{614.9}{6} = 102.5$ lb. per square inch.

The cone must be truly circular in form, and the bracing should be placed as shown in the figure in order to prevent the flue from being pushed down by the pressure exerted on the inclined surface.

Strength of Cone Seam

A tank is 48 in. in diameter and built of 1/4-in. plate which has a tensile strength of 60,000 lb. per square inch. At the lower end of the tank is a cone which has a single-riveted lap seam. The rivet holes are 11/16 in. in diam-

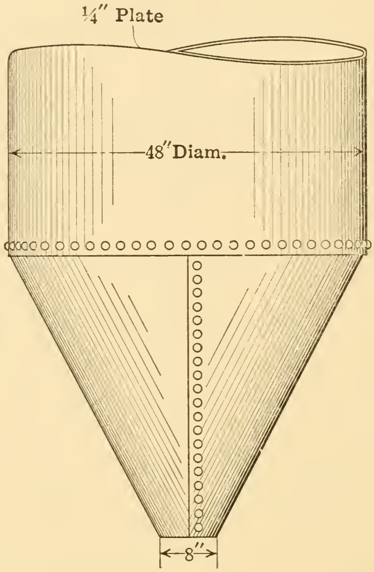


Fig. 17.—Strength of cone seam.

eter and the pitch is 2 in. What is the strength of the seam?

Fig. 17 shows the construction and gives the dimensions.

The longitudinal joint of a cone-shaped section which is withstanding internal pressure is subjected to a vary-

ing stress. The stress at any point in such a joint will be inversely proportional to the distance of the point from the axis of the cone.

It is not practicable to make a joint that would offer equivalent resistance to these varying stresses; therefore it is customary, when the cone is made of one sheet, to make it of the same strength as the tank itself. If several sheets are used in making up the length of the cone, the longitudinal joint of each course might be designed for strength inversely as its distance from the axis. This, however, is rarely done except in large tank work where the size makes the saving in material an object.

The strength of the seam is the strength of the weakest part, which, neglecting the crushing of the sheet in front of the rivets must be either the tensile strength of the ligament between the rivets, or the shearing strength of the rivets themselves.

The strength of the portion of the plate between the rivets is:

$$\frac{2-.6875}{2}\times\frac{60000}{4}=9943.75 \text{ lb. per inch of seam length.}$$

Steel rivets in single shear are assumed to have a shearing strength of 42,000 lb. per square inch of cross-sectional area; the cross-sectional area of a rivet 11/16 in. in diameter is .37122 sq. in., and the strength of the rivets is

$$\frac{.37122\times 42000}{2}=7795.62 \text{ lb.}$$

per inch of seam length, which, being weaker than the ligament, is the strength of the seam.

While the seam is of equal strength throughout its length, the internal pressure per square inch required to shear the rivets will vary directly as the diameter of the cone. For instance, on a line corresponding to a diameter of 40 in. the bursting pressure of the cone will be

$$\frac{7795.62}{20}=389.78 \text{ lb.}$$

per square inch. For a diameter of 16 in. the seam will fail at

$$\frac{7795.62}{8}=974.45 \text{ lb.}$$

per square inch.

Collapsing Pressure of Fire Box

Fig. 18 illustrates a vertical fire-box boiler. It is required to figure the collapsing pressure of the fire box, assuming that 7/8-in. stay bolts are used and that the pressure is 125 lb. per square inch; it is also required to find what pitch to give the rivets in the vertical seam, using 11/16-in. rivets.

In calculating the strength of cylindrical furnaces supported by stay bolts, it is customary to assume that the surface is flat; that is, the tendency of the form to lend strength to the construction is ignored.

With a specified size of bolt the first step is to find how

many square inches of surface one bolt will support at the given pressure of 125 lb. per square inch.

Standard stay bolts up to 1 1/4 in. and larger are cut 12 threads per inch, and it is practically correct to assume that the depth of thread is the same as given by the United States Standard for that pitch, viz., .05425 in., although this is not strictly correct, as the usual stay-bolt thread varies slightly from this standard.

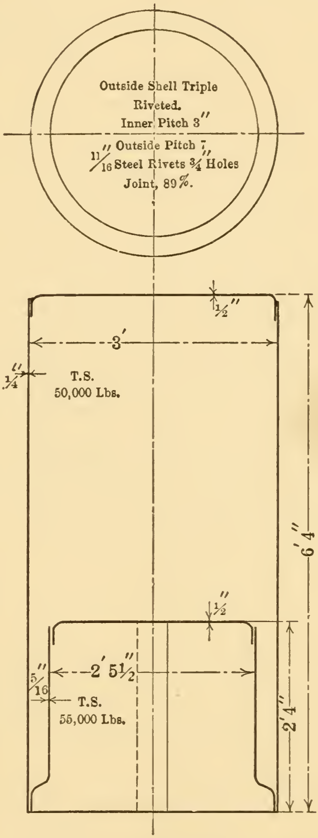


FIG. 18.—Diagram from which to calculate the collapsing pressure of fire box.

Using 7/8-in. bolts will give an effective area of .4614 sq. in. per bolt. The working stress allowed in stay bolts varies, according to different authorities, from 6000 to 10,000 lb. per square inch, the most generally accepted figure being 7500, and on this basis a 7/8-in. stay would support 3460 lb. or at a pressure of 125 lb. per square inch it would be capable of supporting

$$\frac{3460}{125} = 27.6$$

square inches of surface, or the bolts could be spaced

$\sqrt{27.6}$, or about 5 1/4×5 1/4 in. Of course to be strictly correct, the area occupied by the bolt itself should be added to the 27.6 in. before extracting the square root to get the pitch, for the pressure does not act on this area. But in actual practice such allowance is not made.

The surface to be supported in the problem is 2 ft. 6 1/8 in. in diameter, or 94.64 in. in circumference, and the approximate height or distance between the rivets on the leg ring and crown sheet is 24 in. and with the spacing given before it would be necessary to have 18 vertical rows of stays containing four stays each.

There is another feature to be considered in the staying of sheets that affects the strength, and that is the stresses in the sheets themselves. Unwin has devised a formula for the maximum stress in flat sheets supported at regular intervals, the supported points forming squares. The formula is:

$$a=\sqrt{\frac{9ft^2}{2p}}$$

in which a is a side of a square, f the stress per square inch in the sheet, t the thickness of the sheet in inches, and p the pressure per square inch on the supported surface.

Substituting present values in this formula, and assuming a maximum working stress in the sheet of 7500 lb. per square inch, the statement becomes:

$$a=\sqrt{\frac{9\times 7500\times .3125\times .3125}{2\times 125}}=5.13,$$

or 5 1/8 in., which is a little less than the pitch determined by the strength of the stay bolt. The first layout should be modified, making 19 vertical rows of stays with four stays in each row, or the pitch would be about 5×4.8 in.

In relation to pitching the rivets in the vertical seam, it is found in practice that the lapping together of the plate at this point stiffens the sheet so that strength of the joint is not a very important factor in the case, and this joint is generally designed to be least affected by the heat, *i.e.*, single riveted, and of the best proportions to insure tightness, which would be a pitch of about 2 1/4 in. if 5/16-in. plate and 11/16-in. rivets be used.

Relating to Safety-valve Rules

Concerning the apparent discrepancy between the different rules relating to safety-valve calculations, particularly that in Reed's Engineer's Handbook (an English publication), which is used by candidates for British Board of Trade certificates of competency, and those used by a prominent educational institution the reader will discover, after carefully reading the following statements, that both rules referred to will give the same results in any given problem when intelligently used. In other words, both rules are correct, although there is a possibility of misunderstanding them and so obtaining incorrect answers.

If the safety-valve problem could be handled without taking into consideration the effective weight of the lever, then it is not probable that there would ever be any difficulty in candidates having any misunderstanding

of the matter. But the effective weight of the lever *must* be taken into the calculation if accuracy is desired, and of course in such a matter accuracy *should* be desired. Consider what the *effective* weight of a safety-valve lever is and what relation it bears to the calculation.

In the first place there is a difference between the *effective weight* of a lever and the *effective moment* of a lever. Either may be used in the safety-valve problem, but it must be clearly understood how and where each is to be used. Probably the whole difficulty that candi-

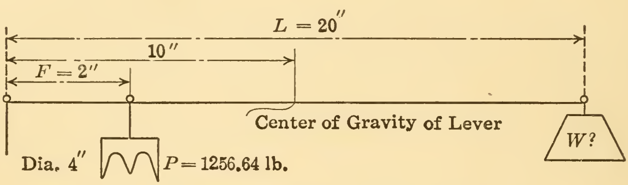


FIG. 19.—Diagram for safety-valve calculations.

dates have with the safety-valve problem lies just in this point. The *effective weight* of a lever is found by multiplying its weight in pounds by the distance its center of gravity is from that point called the fulcrum, and by dividing the product by the distance that the center of the valve upon which the lever acts is from the fulcrum. The *effective moment* of a lever is found by multiplying its weight in pounds by the distance its center of gravity is from the fulcrum. The distances referred to are measured in inches.

Referring to the following example: Required, the weight to be placed at the end of the safety-valve lever shown in the diagram with the given data.

Diameter of valve, 4 in.

Area of valve, 12.5664 sq. in.

Steam pressure per square inch, 100 lb.

Weight of lever, 20 lb.

Weight of valve and stem, 10 lb.

Total upward pressure on the valve is 1256.64 lb. The letter c represents the sum of the effective moments of the valve and stem, and also the lever. The effective *moment* of the lever is $10\times 20=200$; the effective moment of the valve is $2\times 10=20$; and the sum is $200+20=220$. Now, $\frac{FP-c}{L}=$ weight, or $\frac{(2\times 1256.64)-220}{20}=114.664$ lb., the weight to place at the end of the lever. (This method employs effective *moments* of valve and lever.)

Now try the same example by using the other rule, which employs the effective *weight* of the lever, and the *actual* weight of the valve. The rule is stated thus: "To find the weight which must act on a lever at a given distance from the fulcrum so that the valve is about to blow off at a given pressure, subtract the downward force due to the weight of the valve, stem and lever from the product of the area and the steam pressure. Multiply the remainder by the distance from the fulcrum to the center line of the valve, and divide this product by the distance from the fulcrum at which the weight is to act." Expressed in a formula it appears like this:

$$W=\frac{(AP-w)D}{L}$$

$W=$ the required weight in pounds.

$A=$ the area of the valve in square inches.

P = the pressure per square inch.

D = distance in inches from the fulcrum to the center line of valve.

L = distance from fulcrum at which the weight is to be placed.

w = the weight of the valve and stem in pounds plus the effective weight of the lever.

Weight of valve and stem $= 10$ lb.

Effective weight of lever $= \frac{10 \times 20}{2} = 100$ lb.

Total downward force due to both valve and lever = 110 lb.; that is, w in the formula = 110 lb.

$$\frac{(12.5664 \times 100 - 110) \times 2}{20} = 114.654 \text{ lb.},$$

the same answer as obtained by the first method employed.

The *effective weight* of the lever will be $\frac{20 \times 10}{2} = 100$ lb.; that is, this lever, because of its weight and its point of contact with the valve, is equivalent to 100 lb. being placed directly on top of the valve.

The *effective moment* of the same lever will be: $20 \times 10 = 200$.

Now, either the effective weight of the lever or the effective moment may be used in the calculation as has been shown, but each has its own place in the operation, as will be seen a little further on.

The equality of moments in the safety-valve problem are stated like this:

$$W \times D + W' \times D' + w \times d = p \times A \times d$$

In which

W = weight of the weight in pounds.

D = distance in inches that weight is placed from fulcrum.

W' = weight of the lever in pounds.

D' = distance of center of gravity of lever from fulcrum.

w = weight of the valve and stem in pounds.

d = distance center of valve is from the fulcrum.

p = pressure in pounds per square inch.

A = area of valve in square inches.

By applying the formula to the problem it can be seen that the answer obtained is correct. Showing the letters, and substituting the numerals of the example, the statement becomes:

$$W \times D + W' \times D' + w \times d = p \times A \times d,$$
$$114.664 \times 20 + 20 \times 10 + 10 \times 2 = 100 \times 12.5664 \times 2,$$
$$229.328 + 200 + 20 = 2513.28,$$
$$2513.28 = 2513.28.$$

Any safety-valve rule or formula that can successfully stand the above test is correct and may be safely employed.

In order to arrive at the point aimed at, and also to make it as clear as possible, it may be said that from the foregoing formula of moments a series of formulas can be derived that will handle the safety-valve problem in all its phases. Here are the formulas:

(1) $F \times S \div L = W.$

(2) $F \times S \div W = L.$

(3) $L \times W \div S = F$.

(4) $L \times W \div F = S$.

Where F = the force acting upward against the valve; this equals the pressure per square inch times the area of the valve; from the product must be subtracted the weight of the valve and stem.

S = the distance from the fulcrum to the center of the valve.

L = distance from the fulcrum to point where the weight is hung. (This may or may not be at the extreme end of lever.)

W = the weight in pounds of the weight to be hung on lever.

In cases (1) and (2) the *effective weight* of the lever must be subtracted just before *multiplication* occurs. If *effective moment* of lever is to be employed, then the effective moment is to be subtracted just before *division* occurs.

In cases (3) and (4) the *effective weight* of lever is to be added just before *multiplication* occurs.

If the *effective moment* of lever is to be employed, then it will be added just before *division* occurs in the formula. It is very important that a distinction be made between the two phases of the four cases, and then no trouble will be experienced.

Bearing in mind what has just been stated and being ready to refer to it again, now try the problem by using Reed's rule, which is stated thus:

(1) Find the area of the valve and multiply it by the pressure per square inch.

(2) From the product take the weight of the valve (which of course includes the stem).

(3) Multiply the remainder by the distance from the fulcrum to the valve, then subtract the *moment* of the lever, and divide by the distance from the fulcrum to the weight.

In Reed's Engineers' Handbook the *effective moment* of the lever is defined the same as that which appears in the earlier part of this subject.

Using the figures in the example and following Reed's rule, 12.5664 sq. in. (area of valve) $\times$ 100 (pounds pressure per square inch) = 1256.64 lb. (total upward pressure) $-$ 10 lb. (weight of valve and stem) = 1246.64 lb. $\times$ 2 (inches distance from fulcrum to valve) = 2493.28 lb. $-$ 200 (moment of lever) = 2293.28 lb. $\div$ 20 (inches, distance fulcrum to weight) = 114.664 lb. weight. *Ans.*

Thus it is seen that exactly the same answer is obtained as found before. Reed's rule is therefore correct if intelligently used.

Now, suppose that it is desired to use Reed's rule, but instead of having the *effective moment* of lever given, the *effective weight* is given instead. The effective weight of the same lever would be (as before explained) $\frac{20 \times 10}{2}$ = 100 lb. The operation will be like this: 12.5664 sq. in. (area of valve) $\times$ 100 (pounds pressure per square inch) = 1256.64 lb. (total upward pressure against valve) $-$ 10 lb. (weight of valve and stem) = 1246.64 lb. (balance) $-$ 100 lb. (effective weight of lever) = 1146.64 lb. $\times$ 2 (inches, distance, fulcrum to valve) = 2293.28 $\div$ 20 (inches, distance, fulcrum to weight) = 114.664 lb. weight

answer, giving exactly the same answer as in the other two cases. Again is attention directed to the difference between the two latter cases, and particularly the place in each where the subtraction of the lever factor takes place.

Suppose the *effective weight* of the lever were subtracted *after* multiplication had occurred instead of *before*, then the following statement would occur: $12.5664 \times 100 = 1256.64 - 10 = 1246.64 \times 2 = 2493.28 - 100$ (effective weight of lever) $= 2393.28 \div 20 = 119.664$ lb. weight.

$119.664 - 114.664 = 5$ lb. difference, *too much weight.* In other words, by so doing the valve would be over-weighted 5 lb. Of course, the percentage of difference is small, but it is further aggravated by the frictional resistances, and it is not on the safe side of the calculation.

Suppose on the other hand the *effective moment* of the lever were subtracted *before* multiplication occurs; the statement would be: $12.5664 \times 100 = 1256.64 - 10 = 1246.64 - 200$ (effective moment of lever) $= 1046.64 \div 20 = 104.664$ lb. weight.

$114.664 - 104.664 = 10$ lb. difference, too small.

Let the interested reader study this matter out for himself and try the arithmetical operations in the different cases. In this way he cannot but come to a correct and complete understanding of *all* that is embraced in the problem, as far as an operating engineer is concerned, when standing an examination for a certificate.

Any rule relating to the safety valve, which a candidate is not sure of should be tested by the application of the equality of moments formula before referred to.

Roper's Safety-valve Rules

Examiners of engineers in the United States Steamboat Inspection Service sometimes prefer to have candidates for American marine engineers' license use what are known as Roper's Rules for safety-valve problems. Therefore it is well to have a knowledge of these rules in case they are required.

In the following formulas let

A = Area of valve in square inches, or diameter^2 $\times .7854$.

D = distance from center of valve to the fulcrum, in inches.

L = distance of the weight from the fulcrum, in inches.

P = steam pressure in pounds per square inch.

W = weight of the ball in pounds to hang on the lever.

V = weight of the valve and stem in pounds.

w = weight of the lever in pounds.

l = distance of the fulcrum from the center of gravity of the lever, in inches.

$$P = \frac{(W \times L) + (w \times l) + (V \times D)}{A \times D} \qquad (1)$$

$$W = \frac{A \times P \times D - (w \times l + V \times D)}{L} \qquad (2)$$

$$L = \frac{A \times P \times D - (w \times l + V \times D)}{W} \qquad (3)$$

The following examples illustrate the application of the formulas given.

Example 1.—At what pressure will a safety valve blow off, having a diameter of 4 in., weight of valve and stem 12 lb., weight of lever 22 lb., weight of ball 125 lb.; the overall length of lever is 46 in., and it is straight and parallel; the weight is hung at 42 in. from the fulcrum, and the distance from center of valve to fulcrum is 4 in.

Using formula (1) the statement becomes:

$$P=\frac{(125\times 42)+\left(22\times\frac{46}{2}\right)+(12\times 4)}{4^2\times .7854\times 4}$$

$=115.466+$ lb. per square inch, *answer.*

Example 2.—Using the foregoing example (1), it is required to find the necessary weight to hang on the lever.

Using formula (2) the statement becomes:

$$W=\frac{4^2\times .7854\times 115.466\times 4-\left(22\times\frac{46}{2}+12\times 4\right)}{42}$$

$=125$ lb., *answer.*

Example 3.—Using the same example (1), it is required to find at what distance the weight is to be hung.

Using formula (3) the statement becomes:

$$L=\frac{12.5664\times 115.466\times 4-\left(22\times\frac{46}{2}+12\times 4\right)}{125}$$

$=42$ in., *answer.*

Safety-valve Capacity

Extract from paper read before the A. S. M. E., February 23, 1909, by Philip G. Darling, Mechanical Engineer.

CAPACITY FORMULA FOR 45° SEATS

$$1.\quad E = 105 \times l \times P \times D$$

$$2.\quad D = .0095 \frac{E}{l \times P}$$

Modified Forms for Special Applications
For Locomotives

$$3.\quad D = .055 \frac{H}{l \times P}$$

For Cylindrical Multitubular, Vertical and Water Tube Stationary Boilers

$$4.\quad D = .068 \frac{H}{l \times P}$$

For Water Tube Marine and Scotch Marine Boilers

$$5.\quad D = .095 \frac{H}{l \times P}$$

E = pounds of steam relieved per hour.
l = vertical lift of valve in inches.
P = steam pressure (absolute) pounds per square inch.
D = nominal diameter of valve (inlet) in inches.
H = total boiler heating surface in square feet.

For flat-seated valves the constants in these formulæ are as follows: 1—149.; 2—.0067; 3—.052; 4—.065; 5—.090.

UNITED STATES STEAMBOAT INSPECTION SERVICE SAFETY-VALVE RULE

The rule is that the areas shall be found by the formula:

$$A = .2074 \times \frac{WG}{P},$$

in which A = the area of the safety valve in square inches.
W = pounds of water evaporated per square foot of grate surface per hour.
P = the absolute pressure per square inch.
G = grate area in square feet.

In the case of spring-loaded valves, the effective area must be equal to that derived from the formula, and a lever must be provided which will raise the valve one-eighth its diameter from its seat. All seats to have an angle of 45° to the axis of the valves.

Derivation of the United States Board of Supervising Inspectors' Rule for Areas of Safety Valves

Napier's rule for flow of steam through orifices:

$$\text{Flow in pounds per second} = \frac{\text{Absolute pressure} \times \text{area}}{70}$$

(This corroborated by Peabody's experiments.)
P = absolute pressure = gage pressure + 15.
W = pounds discharged per hour.
A = area of valve opening or orifice.

Hence

$$W = \frac{P \times A}{70} \times \overset{sec.}{60} \times \overset{min.}{60} = \frac{360 \times A \times P}{7}$$

For safety-valve practice, cut this amount down 25 per cent., leaving 75 per cent.
Thus

$$W = .75 \times \frac{360}{7} \times A \times P = \frac{270 \times A \times P}{7}$$

Restrict the lift of valve to 1/32 of its diameter $= \dfrac{d}{32}$

then

$$A \frac{d}{32} \times \pi \times d = \text{lift} \times \text{circumference} = \frac{\pi \times d^2}{32}$$

Substituting this value for A = area of orifices

$$W = \frac{270}{7} \times P \times \frac{\pi d^2}{32}$$

In a valve of diameter d the area =

$$\frac{\pi d^2}{4} = a$$

To get W in terms of area of valve, substitute for d^2 its value in terms of a,

$$d^2 = \frac{4a}{\pi}$$

$$W = \frac{270}{7} \times P \times \frac{\pi}{32} \times \frac{4a}{\pi} = 4.821 \times Pa$$

In safety-valve practice this will represent the pounds

of steam that must escape per hour, which must be equal to the pounds of water that the boiler can evaporate per hour.

To reduce this to a working basis, consider these quantities per square foot of grate surface per hour.

W = pounds of water evaporated per square foot of grate surface per hour.

P = absolute pressure per square inch.

A = area of safety valve per square foot of grate surface.

Hence

$$W = 4.821 \times P \times a, \text{ and } a = .2074 \times \frac{W}{P}$$

From which a table of areas required per square foot of grate surface may be found by assuming the different values of W and P.

Finding the Center of Gravity of Tapered Safety-valve Levers

In questions relating to the lever safety valve it is necessary to know how to find the center of gravity of the lever, in order to calculate the effective weight of the lever; how may the center of gravity be found when the lever is tapered, and of uniform thickness throughout its entire length?

Answer.—There are three ways that the center of gravity of tapered safety-valve levers is found: By taking the lever off and actually balancing it on a knife edge; diagrammatically, and mathematically. The last two methods only require an explanation.

First consider the diagrammatic method of finding the center of gravity. To be brief and to the point, assume a lever 20 in. long, 2 in. wide at one end, and 1 in. wide at the other end and of uniform thickness throughout. Do not consider the projection which is usually at the small end of the lever for preventing the weight slipping off, nor the holes at the other end through which the lever is attached to the fulcrum, and by which the pin is attached which bears upon the valve. These would

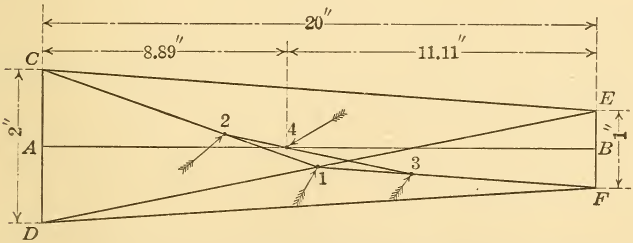


FIG. 20.—Diagram for finding center of gravity of tapered lever.

make but little difference in the result obtained, and would tend to complicate the calculations.

An inspection and study of figure 20 should make clear how the center of gravity may be determined. A scale of quarter size may be chosen as a matter of convenience only. Any other scale will do. The center line *AB* is drawn first. The length of *AB* is 5 in., or one-quarter as long as the lever assumed. At *A* and *B* respectively draw the lines *CD* and *EF* 1/2 in. and 1/4 in., representing the actual 2-in. and 1-in. dimensions of the lever.

Next draw the diagonal line *DE*, and locate the point 1 exactly midway between points *D* and *E*. Join the

points C and 1 by the line $C1$. From the point 1 locate the point 2 on the line $C1$ at a distance equal to one-third of its length. Now, from the point F draw the line $F1$, and from the point 1 locate the point 3 on the line $F1$, one-third of its length from the point 1. Join points 2 and 3; where the line 2–3 intersects the center line AB will be the point 4 at which the center of gravity is. By carefully and accurately drawing the figure the measurements show that the center of gravity is about 11.1 in. from the small end of the lever, and is about 8.9 in. from the large end of the lever.

To find the center of gravity of a tapered lever mathematically, the following formula may be used:

$$x = \frac{2A+B}{3A+3B} \times L; \text{ or in this form, } x = \frac{2A+B}{3(A+B)} \times L$$

Where x = distance center of gravity is from the small end of lever.
A = the width in inches of lever at the fulcrum end.
B = width in inches of lever at the weight end.
L = entire length of lever in inches.

Applying this formula to our example we have

$$\frac{2 \times 2+1}{3(2+1)} \times 20 = \frac{5}{9} \times 20 = \frac{100}{9} = 11.11 \text{ in.,}$$

distance center of gravity from small end of lever.

Thus it will be seen that the two methods which have just been explained verify each other, and no doubt the first method (that of balancing the lever on a knife edge) referred to would verify the others, for, as before

stated, the difference due to the projection at one end and the holes at the other end would hardly be noticeable.

At an examination where time is limited, it would be better to use the formula in such questions. It does not require much study and only a little practice to become familiar with it. Furthermore, it will be found easy to remember after it has been used a few times. Marine engineers cannot afford to ignore this subject.

Chimneys

The "proportions of chimneys" vary very much according to the requirements. Every chimney should be large enough in cross-section to carry off the gases and high enough to produce sufficient draft to cause a rapid combustion. The object of a chimney being to carry off the waste gases, it naturally determines the amount of fuel that can be burnt per hour, and it is advisable to have always a good draft, as it can always be regulated by a damper.

Draft pressure is caused by the difference in weight between a column of hot gases in the chimney and a column of air of equal height and area outside the chimney.

Formula for finding the force of draft in inches of water of any given chimney:

$$F = H\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right)$$

Where F = force of draft in inches of water.
H = height of chimney in feet.

T_1 = absolute temperature of chimney gases $(t+460)$.

T_2 = absolute temperature of the external air (t_1+460).

t = temperature of chimney gases.

t_1 = temperature of external air.

Formula for finding the height of a chimney in feet for a given force of draft:

$$H = \frac{F}{\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right)}$$

To find the maximum force of draft for any given chimney, the external air being 60° F., and the heated column being 600° F., multiply the height above the grate in feet by .0073, and the product is the force of draft expressed in inches of water.

1. The draught power of the chimney varies as the square root of the height.

2. The retarding of the ascending gases by friction may be considered as equivalent to a diminution of the area of the chimney, or to a lining of the chimney by a layer of gas which has no velocity. The thickness of this lining is assumed to be 2 in. for all chimneys, or the diminution of area equal to the perimeter × 2 in. (neglecting the overlapping of the corners of the lining). Let D = diameter in feet, A = area, and E = effective area in square feet.

For square chimneys, $E = D^2 - \frac{8D}{12} = A - \frac{2}{3}\sqrt{A}$

For round chimneys, $E = \frac{\pi}{4}\left(D^2 - \frac{8D}{12}\right) = A - .591\sqrt{A}$

For simplifying calculations, the coefficient of $\sqrt{A}$ may be taken as .6 for both square and round chimneys, and the formula becomes

$$E = A - .6\sqrt{A}$$

3. The power varies directly as this effective area E.

4. A chimney should be proportioned so as to be capable of giving sufficient draught to cause the boiler to develop much more than its rated power, in case of emergencies, or to cause the combustion of 5 lb. of fuel per rated horse-power of boiler per hour.

5. The power of the chimney varying directly as the effective area E, and as the square root of the height H, the formula for horse-power of boiler for a given size of chimney will take the form horse-power $= CE\sqrt{H}$, in which C is a constant, the average value of which, obtained by plotting the results obtained from numerous examples in practice, the author finds to be 3.33.

The formula for horse-power then is

$$\text{horse-power} = 3.33E\sqrt{H}, \quad \text{or} \quad \text{horse-power} = 3.33\ (A - .6\sqrt{A})\sqrt{H}$$

If the horse-power of boiler is given, to find the size of chimney, the height being assumed,

$$E = \frac{.3\ H.P.}{\sqrt{H}};\ = A - .6\sqrt{A}$$

For round chimneys, diameter of chimney = diam. of $E + 4$ in.

For square chimneys, side of chimney $= \sqrt{E} + 4$ in.

If effective area E is taken in square feet, the diameter in inches is $d = 13.54\sqrt{E} + 4$ in., and the side of a square chimney in inches is $s = 12\sqrt{E} + 4$ in.

If horse-power is given and area assumed the height $H = \left(\frac{0.3\ H.P.}{E}\right)^2$.

In proportioning chimneys the height is generally first assumed, with due consideration to the heights of surrounding buildings or hills near to the proposed chimney, the length of horizontal flues, the character of coal to be used, etc., and then the diameter required for the assumed height and horse-power is calculated by the formula or taken from the table.

Size of Boiler Feed Pipe

What size feed pipe should be installed to supply three 100-H.P. boilers?

At ordinary commercial rating the water required per horse-power-hour, is taken as 30 lb. Each boiler therefore will need 3000 lb. of water per hour or 50 lb. per minute, which at 62.5 lb. per cubic foot would call for .8 of a cubic foot each minute.

To provide for emergencies, twice the actual quantity of water required should be figured. The velocity of flow is usually taken as 100 ft. per minute. On this basis, the quantity to be taken care of will be $2 \times .8 = 1.6$ cu. ft. per minute, which at 100 ft. per minute velocity would require an area of pipe of $\frac{1.6}{100} = .016$ sq. ft., or 2.3 sq. in., found like this:

$.016 \times 144 = 2.304$ sq. in. and $\sqrt{\frac{2.304}{.7854}} = 1.7$ in. diam.

The nearest commercial size of pipe is 1 1/2 in. diameter, which is the size to connect to each of the boilers.

For the main pipe, to supply all three boilers, a pipe of equivalent area to the three boiler pipes would be 6.9 square inches, and this would mean a 3-in. diameter pipe is required. Should not all three boilers require to be fed at the rate before referred to at any one time, then a 2 1/2-in. main feed pipe will be ample.

Effect of Stiffness of Head on Braces

Away back in 1876, Samuel Nichols, a practical boiler maker in charge of a large English works, wrote a book for boiler makers. In it he recounts some experiments made under the supervision of Robert Nelson, author of "A Treatise on Steam Boilers," upon boilers built by himself for the purpose, with regard to unstayed heads in cylindrical shells. A boiler 30 in. in diameter with flat heads 3/8 in. thick, of plate having a "tenacity" of 21.2 tons per square inch with an elongation of 7.9 per cent., flanged on a radius of 1 in. inside the plate, was subjected to hydrostatic pressure. At 10 lb. the head had bulged 1/16, at 120 lb. 11/16 and at 150 lb. 13/16 in. Permanent set occurred somewhere between 50 and 65 lb. at which latter figure the deflection was 3/8 in. Rupture took place at about 300 lb.

From the results of his tests, Mr. Nichols deduced a formula which is printed in his book for determining the bursting pressure of cylinders with unstayed heads, although he strongly disclaimed any advocacy of exposing an unstayed surface to high pressures. "On the

contrary," he writes of himself, "he is more convinced, now that he has witnessed these experiments, that a flat unstayed surface is very weak indeed, and that they still require a larger amount of care and judgment on the part of boiler engineers than any portion of the boiler."

Evidently under the erroneous notion that if the head can take care of the amount of pressure indicated by this formula on its own account it will need correspondingly less bracing, the Board of Boiler Rules of Ohio, after instructing inspectors to determine the working pressure of boilers with respect to the bracing in the usual way but with an allowable stress of 8000 lb. per square inch irrespective of size, tells them, "To the above pressure may be added the Nichols formula with a factor of safety of not less than 8."

If this means, and it can seem to mean nothing else, that to the pressure which can safely be taken care of by the bracing may be added the pressure which, by the Nichols formula, would be allowed upon the unbraced head, it is wrong.

Mr. Nichols shows that the head may be bulged considerably without straining the sheet beyond the elastic limit. But a brace is supposed to be tight before the head commences to bulge. Just as soon as the head starts to move it commences to stretch the brace. Under the allowable stress of 8000 lb. per square inch of section a brace will extend only about 1/3750 of its length, or a 6-ft. brace would extend less than .02 in. The pressure which would produce the movement in the unbraced head is inconsiderable—a pressure of 10 lb. produced a movement of three times as much—and yet this is

all the help that the stiffness of the head would be to the brace.

As an example, assume a 72-in. boiler, height of segment to be braced 24 in., area of segment to be braced 814 sq. in., pressure 100 lb., thickness of head 1/2 in., tensile strength 60,000 lb.

The Nichols formula with a factor of safety of 8 would allow

$$\frac{t \times T \times 10}{A \times 8} = \frac{0.5 \times 60000 \times 10}{814 \times 8} = 46 \text{ lb.}$$

If this may be added to the pressure which the bracing is capable of carrying, it would be necessary to brace against only $100 - 46 = 54$ lb. per square inch, which at 8000 lb. per square inch would require seven 1-in. braces. Practice calls for at least twice as many.

Bracing Flat Surfaces in Steam Boilers

There is considerable variation as to the load allowed per square inch of net section on diagonal braces, rod braces and stay bolts by the authorities who have laid down rules on this subject. The United States Government rules allow 6000 lb. on welded iron stays below 1 1/4 in., 7500 on 1 1/4 in. and above, and from 7000 to 9000 lb. on weldless steel stays. Chicago has a flat scale of 6000 lb. on all stays or braces, Philadelphia has a limit of 7500 lb.; the Massachusetts rules allow from 6500 to 9000 lb. per square inch net section, varying as the braces are welded or weldless and with the size, the latter for the reason that with a given waste of

material the percentage of reduction is greater with the smaller sizes.

The above applies to flat surfaces and refers to flat heads, such as dome heads, segments of heads, etc. The United States Government has a rule to find the pressure on flat heads not exceeding 20 in. in diameter as follows:

$$P = \frac{C \times T^2}{A}$$

where P = pressure.

C = 112 (7/16 or under) and 120 over 7/16.

A = 1/2 the area.

T = thickness in sixteenths.

With a 3/4-in. head 20 in. in diameter, 110 lb. would be allowed by these rules.

A short time ago the Board of Boiler Rules for the State of Ohio issued instructions to the inspectors holding certificates of competency that the following formula could be used in flat surface of heads:

$$P = \frac{T \times T.S. \times 10}{A \times 8}$$

where T = thickness.

$T.S.$ = tensile strength.

A = area.

In addition, a limitation allowance of 8000 lb. per square inch irrespective of size of brace is granted. This applies to boilers now in use, but not to boilers to be installed after July 1, 1912. This ruling is far more liberal than any other authority has heretofore

allowed as a comparison will show. Assume a 72-in. boiler, height of segment 24 in., thickness of head 1/2 in., tensile strength 60,000 lb. The total area of the segment = 1186.4 sq. in., while the area requiring bracing = 814 sq. in. Hence

$$\frac{.5 \times 60000 \times 10}{814 \times 8} = 46 \text{ lb.}$$

allowed without braces.

Let the pressure required be equal to 100, then 100 − 46 = 54 lb. to be braced, and 54 × 814 = 43,956 lb. Assuming the proposed brace to be of .79 in. area, then .79 × 8000 = 6320 per brace, and 43,956 ÷ 6320 = 7 braces of practically 1-in. diameter.

It may be said that flat surfaces subjected to internal pressure will spring and proportionally to the unsupported area. Samuel Nichols, in his tests of circular flat heads, showed the springing began with very low pressures, even at 20 lb. on 28-in. heads and increased as the pressure was raised. Applying this fact then to the Ohio ruling, it seems the head would so spring that at 100 lb. pressure the total load on the braces would be .79 × 7 = 5.53 into the toal load, 81,400 lb., or 14,718 per square inch net section instead of 8000 lb. Applying this ruling to a flat dome head 36 in. diameter, 1/2 in. thick, 60,000 lb. tensile strength, area to be braced 707 in., gives 53 lb. without bracing. The results of allowing a flat head unbraced to spring and return times without number would be final failure due to such action.

Reverting to the segment as in a horizontal tubular boiler, it may be said other authorities have been careful

to avoid allowing excessive stresses on the chord of the segment which is supported by the tubes inasmuch as the latter are not a constant in strength as is the flange of the head in the arc of the segment, and this view has been approved by most students as the tubes are subjected to more or less rapid wear and reduction in thickness.

Further, such calculations apply to boilers now in use irrespective of age. Indeed, Ohio has no limitation as to age as respects pressure to be determined with a factor of safety of 4 together with this exceedingly liberal allowance on braces. Comparing this with the Massachusetts, Chicago, Philadelphia and Detroit rules, what results may be expected?

Three Boiler Questions

In an examination, three out of five engineers failed to answer the following questions, which are given for the benefit of those who may be called upon to make similar calculations.

1. A horizontal tubular boiler is 72 in. in diameter and 18 ft. long; thickness of plate .437 in.; efficiency of longitudinal joints 77 per cent., and steam pressure 110 lb. What should be the tensile strength of the plate, allowing a factor of safety of 5?

2. If the tensile strength had been 56,000 lb. and the efficiency of the joint 70 per cent., what thickness of plate should be employed?

3. If this boiler had been intended for 125 lb. of steam, what would the efficiency of the joint have been, using the data in the first question (except the pressure and efficiency)?

The required tensile strength of plate is found by the rule

$$TS = \frac{\text{pressure} \times \text{diam.} \times \text{factor of safety}}{\text{efficiency of joint} \times \text{thickness of plate} \times 2}$$

Substituting the figures given, instead of the words in the rule, we have

$$\text{Tensile strength} = \frac{110 \times 72 \times 5}{.77 \times 437 \times 2} = 58{,}775 \text{ lb.}$$

The rule for thickness of plate is

$$\text{Thickness} = \frac{\text{pressure} \times \text{diam.} \times \text{factor of safety}}{\text{tensile strength} \times \text{efficiency of joint} \times 2}$$

Again substituting the figures we have

$$\text{Thickness} = \frac{110 \times 72 \times 5}{56000 \times .70 \times 2} = .5 \text{ in.}$$

Efficiency of joint is given by the rule

$$\text{Efficiency} = \frac{\text{pressure} \times \text{diam.} \times \text{factor of safety}}{\text{tensile strength} \times \text{thickness} \times 2}$$

This figured out gives

$$\text{Efficiency} = \frac{125 \times 72 \times 5}{58775 \times .437 \times 2} = .875 \text{ or } 87.5 \text{ per cent.}$$

To Find Pitch of Rivets

How can the pitch of the rivets be determined for a double-riveted butt and double-strap joint which is to have 7/8-in. rivets and a strength of plate between the rivet holes on the outer row which will be 82 per cent. of the strength of the solid plate?

Let P = pitch.

t = thickness of plate.

TS = tensile strength of plate.

d = diameter of rivets.

Then

$P \times t \times TS =$ strength of solid plate

and

$(P-d) \times t \times TS =$ strength of plate between the rivet holes on the outer row. The conditions require

$$\frac{(P-d)t \times TS}{P \times t \times TS} = .82 \qquad (1)$$

By canceling out t and TS from numerator and denominator of the first member of (1) we obtain,

$$\frac{P-d}{P} = .82 \qquad (2)$$

and as $d = .875$, the equation becomes

$$\frac{P-.875}{P} = .82 \qquad (3)$$

from which it is found that

$$P = \frac{.875}{.18} = 4.86+ \qquad (4)$$

or practically 4 7/8 in.

Collapsing Pressure of Lap-welded Bessemer Steel Tubes of from 3 to 10 In. Diameter, and of Different Wall Thicknesses

Formulas:

$$p = 86,670\frac{t}{d} - 1386$$

$$p = 1000\left(1 - \sqrt{1 - 1600\frac{t^2}{d^2}}\right)$$

and

$$p = 50,210,000\left(\frac{t}{d}\right),$$

where p = collapsing pressure in pounds per square inch.
d = outside diameter of tube in inches.
t = wall thickness in inch measure.

The first formula is applicable to cases where $\frac{t}{d}$ is greater than .023 and the others to the case of thin-walled tubes where the quotient is less than that value.

Safe Working Pressure Calculations as Applied to the Shell of Climax, Hazelton and Porcupine Types of Steam Boilers

Example.—Shell plate 5/8 in. thick, diameter 30 in., tensile strength 60,000 lb. per square inch, tubes 4 in. diameter. See Figs. 1 and 2 of this section.

Consider the ring of the shell 3 27/32 in. wide (Fig. 2) included between any two transverse rows of holes. For each pound per square inch of pressure, any longitudinal section of this ring will be subjected to a stress of $\frac{30 \times 3\ 27/32}{2} = 57.65625$ lb. The net section of the ring on the axis *ab* of a longitudinal row of holes is $5/8 \times (3\ 27/32 - 2) = 1.15+$ sq. in. The unit stress on this section for a pressure of 1 lb. per square inch is, therefore, $57.656 \div 1.15 = 50$ lb. per square inch, nearly. The section on the line *ac* through two adjacent holes in a diagonal row is subjected to the stress of 57.656 lb., which acts in the direction of the line *ef*, perpendicular to *ab*. This stress may be resolved into two components, one of which, *eg*, acts perpendicular to *ac* and tends to pull the plate apart through that section, while the other, *eh*, acts along the line *ac* and tends to shear the plate through the

same section. Of these two components, *eg* is equal to the stress on a longitudinal section of the ring multiplied by the cosine of the angle *bac*, and *eh* is equal to the stress on the longitudinal section multiplied by the sine of the angle *bac*. The sine of *bac* is $3.5 \div 5.2 = .673$, and the cosine $3\ 27/32 \div 5.2 = .739$, which correspond to an angle of 47° 40′, nearly. The tensile stress on the section *ac* resulting from the stress of 57.656 lb. acting perpendicular to the section *ab* is, therefore, $57.656 \times .739 =$

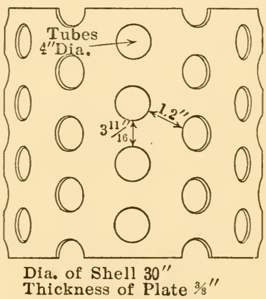


FIG. 1.—Calculations relating to porcupine type of boilers.

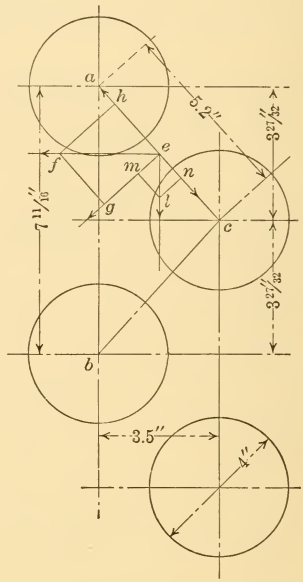


FIG. 2.—Calculations relating to porcupine type of boilers.

42.61 lb., nearly, and the shearing stress is $57.656 \times .673 =$ 38.8 lb. In addition to the stresses due to the force that tends to break the ring through a longitudinal section, the section on *ac* is subjected to a stress from the action of the force that tends to rupture the shell along a transverse section. For a pressure of 1 lb. per square inch,

this force is equal to the area of the head multiplied by 1; that is, to $30^2 \times .7854 \times 1 = 706.86$ lb. The number of sections among which this force is divided is equal to the circumference of a 30-in. circle divided by 3.5; that is, to $\frac{30 \times 3.1416}{3.5} = 27$. The force on each section is, therefore, $706.86 \div 27 = 26.18$ lb. This force acts on the section *ac* in the direction of the line *el*, parallel to the line *ab*. It may be resolved into two components, one, *em*, perpendicular to *ac*, which is equal to 26.18 multiplied by the sine of the angle *bac;* the other, *en*, in the direction of *ac*, equal to 26.18 multiplied by the cosine of *bac*. Of these two components, the first acts in the same direction as the component *eg;* its value, $26.18 \times .673 = 17.62$ lb., nearly, is, therefore, to be added to the value represented by *eg*, thus giving us a total tensile stress in the section *ac* of $42.61 + 17.62 = 60.23$ lb. The component *en*, whose value is $26.18 \times .739 = 19.347$ lb., acts in the opposite direction to *eh;* they therefore partly neutralize each other, and the resulting shearing stress, $38.8 - 19.347 = 18.453$, is so much less than the tensile stress of 60.23 lb. that it is evident the section would fail by tension and not by shear. The area of the net section of the plate, which resists the tensile stress of 60.23 lb., is $1.2 \times 5/8 = .75$ sq. in.; the unit stress in this section for a pressure of 1 lb. per square inch is, therefore, $60.23 \div .75 = 80.3$ lb. per square inch. Since the stress on the section *ab* was but 50 lb. per square inch, it is evident that the plate will fail along the section *ac*. If we assume the safe working stress of the 60,000-lb. steel plate to be 10,000 lb. per square inch, the safe working

pressure will be $10,000 \div 80.3 = 124.5$ lb. per square inch.

Figuring the Safe Working Pressure of the Shell of a Locomotive Boiler

How to determine the safe working pressure of the shell of a locomotive boiler with several courses of varying diameter, like that shown in Fig. 21.

With the locomotive boiler, like Fig. 21, the safe working pressure can only be ascertained by considering the

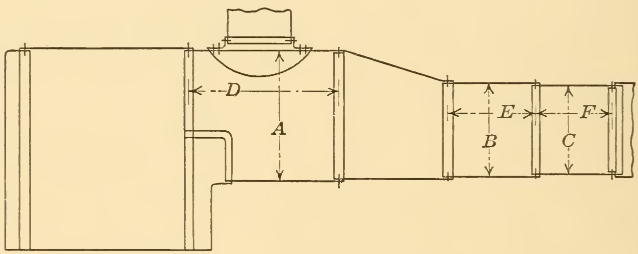


Fig. 21.—Diagram from which to calculate the safe pressure in a locomotive boiler.

diameters *A*, *B* and *C*. Also, the thickness of the plates and the efficiencies of the longitudinal seams of the respective courses must be considered.

The inside diameter *A* is 72 in. and plate 3/4 in. in thickness, the inside diameter *B* 61 in. and plate 9/16 in. in thickness, and the inside diameter *C* 60 in. and plate 1/2 in. in thickness. The efficiency of the riveted joints for the respective courses is *A*, 82 per cent., course *B* 84 per cent., and course *C* 86 per cent.

It may be asked, what is the difference in efficiency in

the respective courses? This is because the over-all distances, D, E and F, are such that the same maximum pitch could not be obtained in the respective courses, and a change of pitch, large or small, will make a difference in the efficiency of the net section of plate, maximum pitch of rivets, which point is made the weaker of the several parts of a riveted joint.

Assuming the factor of safety to be 5, and the plate to have a tensile strength of 60,000 lb. then the working pressure of the boiler, as far as the shell is concerned, may be determined by the following formula:

Where T = thickness of the plate in inches.
D = diameter of the boiler in inches.
T_s = tensile strength of the plate in pounds.
F = factor of safety.
E = efficiency of the longitudinal seam.
P = pressure in pounds per square inch.

$$\frac{T \times T_s \times E}{D \times F} = P$$

The safe working pressure for course A then will be

$$\frac{(2 \times 3/4) \times 60000 \times .82}{72 \times 5} = 205 \text{ lb.}$$

The safe working pressure for course B then will be

$$\frac{(2 \times 9/16) \times 60000 \times .84}{61 \times 5} = 186 \text{ lb.}$$

The safe working pressure for course C then would be

$$\frac{(2 \times 1/2) \times 60000 \times .86}{60 \times 5} = 172 \text{ lb.}$$

The calculations thus show that the course *C*, the course with the least diameter and the longitudinal seam with the greatest efficiency, to be the weaker of the three courses *A*, *B* and *C*. Therefore, the pressure for the boiler, considering the shell only, would be 172 lb.

Had the designer in the first instance made the course *C* 9/16 plate and the course *B* 5/8-in. plate, the boiler shell in question could have been allowed a greater pressure than 172 lb. To a boiler designer these calculations would suggest several things. First would be that if no change was to be made in course *C*, then the thickness of course *A* could be reduced, perhaps, 1/16 in. in thickness. This would make a saving in the cost of the boiler. Second, if need be the efficiencies of the longitudinal seams of courses *A* and *B* could be less. This is, of course, on the assumption that course *A* will undergo no changes in regard to thickness of plate, diameter, efficiency of longitudinal seam and tensile strength of plate.

PART III

APPENDIX

EXTRACTS FROM UNITED STATES RULES—MARINE—AND FROM THE BOARD OF BOILER RULES STATE OF MASSACHUSETTS—TABLES

APPENDIX

EXTRACTS FROM RULES OF THE UNITED STATES BOARD OF SUPERVISING INSPECTORS, STEAMBOAT INSPECTION SERVICE

United States Rules Pertaining to Riveted Joints

The following formulas, equivalent to those of the British Board of Trade, are given for the determination of the pitch, distance between rows of rivets, diagonal pitch, maximum pitch, and distance from centers of rivets to edge of lap of single- and double-riveted lap joints, for both iron and steel boilers.

Let p = greatest pitch of rivets in inches.
n = number of rivets in one pitch.
p_d = diagonal pitch in inches.
d = diameter of rivets in inches.
T = thickness of plate in inches.
V = distance between rows of rivets in inches.
E = distance from edge of plate to center of rivet in inches.

TO DETERMINE THE PITCH

Iron plates and iron rivets:

$$p = \frac{d^2 \times .7854 \times n}{T} + d$$

Example, first, for single-riveted joint: Given, thickness of plate $(T) = 1/2$ in., diameter of rivet $(d) = 7/8$ in. In this case $n = 1$. Required the pitch.

Substituting in formula, and performing operation indicated,

$$\text{Pitch} = \frac{(7/8)^2 \times .7854 \times 1}{1/2} + 7/8 = 2.077 \text{ in.}$$

Example for double-riveted joint: Given, $t = 1/2$ in., and $d = 13/16$ in. In this case, $n = 2$. Then—

$$\text{Pitch} = \frac{(13/16)^2 \times .7854 \times 2}{1/2} + 13/16 = 2.886 \text{ in.}$$

For *steel* plates and steel rivets:

$$p = \frac{23 \times d^2 \times .7854 \times n}{28 \times T} + d$$

Example for single-riveted joint: Given, thickness of plate = 1/2 in., diameter of rivet 15/16. In this case, $n = 1$.

$$\text{Pitch} = \frac{23 \times (15/16)^2 \times .7854 \times 1}{28 \times 1/2} + 15/16 = 2.071 \text{ in.}$$

Example for double-riveted joint: Given, thickness of plate = 1/2 in., diameter of rivet = 7/8 in. $n = 2$. Then—

$$\text{Pitch} = \frac{23 \times (7/8)^2 \times .7854 \times 2}{28 \times 1/2} + 7/8 = 2.85 \text{ in.}$$

FOR DISTANCE FROM CENTER OF RIVET TO EDGE OF LAP

$$E = \frac{3 \times d}{2}$$

Example.—Given, diameter of rivet $(d) = 7/8$ in., required the distance from center of rivet to edge of plate.

$E = \dfrac{3 \times 7/8}{2} = 1.312$ in., for single- or double-riveted lap joint.

FOR DISTANCE BETWEEN ROWS OF RIVETS

The distance between lines of centers of rows of rivets for double, chain-riveted joints (V) should not be less than twice the diameter of rivet, but it is more desirable that V should not be less than $\frac{4d+1}{2}$.

Example under latter formula: Given, diameter of rivet = 7/8 in.; then—

$$V = \frac{(4 \times 7/8) + 1}{2} = 2.25 \text{ in.}$$

For ordinary, double, zigzag riveted joints:

$$V = \frac{\sqrt{(11p + 4d)(p + 4d)}}{10}.$$

Example.—Given, pitch = 2.85 in. and diameter of rivet = 7/8 in.; then—[1]

$$V = \frac{\sqrt{(11 \times 2.85 + 4 \times 7/8)(2.85 + 4 \times 7/8)}}{10} = 1.487 \text{ in.}$$

DIAGONAL PITCH

For double, zigzag riveted lap joint. Iron and steel:

$$p_d = \frac{6p + 4d}{10}$$

Example.—Given, pitch = 2.85 in., and d = 7/8 in.; then—

$$p\,d = \frac{(6 \times 2.85) + (4 \times 7/8)}{10} = 2.06 \text{ in.}$$

[1] Extract the square root of the expression above the line only, then divide by 10.

Maximum Pitches for Riveted Lap Joints

For single-riveted lap joints:

$$\text{Maximum pitch} = (1.31 \times T) + 1\ 5/8.$$

For double-riveted lap joints:

$$\text{Maximum pitch} = (2.62 \times T) + 1\ 5/8.$$

Example.—Given, a thickness of plate = 1/2 in., required the maximum pitch allowable.

For single-riveted lap joint:

$$\text{Maximum pitch} = (1.31 \times 1/2) + 1\ 5/8 = 2.28 \text{ in.}$$

For double-riveted lap joint:

$$\text{Maximum pitch} = (2.62 \times 1/2) + 1\ 5/8 = 2.935 \text{ in.}$$

TO DETERMINE THE AREAS OF DIAGONAL STAYS

Multiply the area of a direct stay required to support the surface by the slant or diagonal length of the stay; divide this product by the length of a line drawn at right angles to surface supported to center of palm of diagonal stay. The quotient will be the required area of the diagonal stay.

$$A = \frac{a \times L}{l}$$

Where A = sectional area of diagonal stay.
a = sectional area of direct stay.
L = length of diagonal stay.
l = length of line drawn at right angles to boiler head or surface supported to center of palm of diagonal stay.

Given diameter of direct stay = 1 in., $a = .7845$, $L = 60$ in., $l = 48$ in., substituting and solving,

$$A = \frac{.7854 \times 60}{48} = .981 \text{ sectional area.}$$

Diameter = 1.11 in. = 1 1/8 in.

The sectional area of gusset stays, when constructed of triangular right-angled web plates secured to single or double angle bars along the two sides at right angles, shall be determined by formula for diagonal stays, and shall not be less than 10 per cent. greater than would be necessary for a diagonal bolt stay.

STAYS

The maximum stress in pounds allowable per square inch of cross-sectional area for stays used in the construction of marine boilers, when same are accurately fitted and properly secured, shall be ascertained by the following formula:

$$P = \frac{A \times C}{a}$$

Where P = working pressure in pounds.
A = least cross-sectional area of stay in inches.
a = area of surface supported by one stay in inches.
C = a constant.
C = 9000 for tested steel stays 1 1/4 in. and upward in diameter, when such stays are not forged or welded. The ends may be upset to a

sufficient diameter to allow for the depth of the thread, provided it is the least diameter of the stay. All such stays after being upset shall be thoroughly annealed.

$C = 8000$ for a tested Huston or similar type of brace, the cross-sectional area of which exceeds 5 sq. in.

$C = 7000$ for such tested braces when the cross-sectional area is not less than 1.227 and not more than 5 sq. in., provided such braces are prepared at one heat from a solid piece of plate without welds.

$C = 7500$ for wrought iron through stays 1 1/4 in. diameter and upward. When made of the best quality of refined iron, they may be welded.

$C = 6000$ for welded crowfoot stays when made of the best quality of refined wrought iron, and for all stays not otherwise provided for when made of the best quality of refined iron or steel without welds.

Furnaces

The tensile strength of steel used in the construction of corrugated or ribbed furnaces shall not exceed 67,000 and be not less than 54,000 lb.; and in all other furnaces the minimum tensile strength shall not be less than 58,000 and the maximum not more than 67,000 lb. The minimum elongation in 8 in. shall be 20 per cent.

All corrugated furnaces having plain parts at the ends

not exceeding 9 in. in length (except flues especially provided for) when new, and made to practically true circles, shall be allowed a steam pressure in accordance with the following formula:

$$P = \frac{C \times T}{D}$$

LEEDS SUSPENSION BULB FURNACE

$$P = \frac{C \times T}{D}$$

Where P = pressure in pounds.
T = thickness in inches, not less than 5/16 in.
D = mean diameter in inches.
C = a constant, 17,300, determined from an actual destructive test under the supervision of the Board, when corrugations are not more than 8 in. from center to center, and not less than 2 1/4 in. deep.

MORISON CORRUGATED TYPE

$$P = \frac{C \times T}{D}$$

Where P = pressure in pounds.
T = thickness in inches, not less than 5/16 in.
D = mean diameter in inches.
C = 15,600, a constant, determined from an actual destructive test under the supervision of the Board of Supervising Inspectors,

when corrugations are not more than 8 in. from center to center and the radius of the outer corrugations is not more than one-half of the suspension curve.

[In calculating the mean diameter of the Morison furnace, the least inside diameter plus 2 in. may be taken as the mean diameter, thus—

Mean diameter = least inside diameter + 2 in.]

FOX TYPE

$$P=\frac{C\times T}{D}$$

Where P = pressure in pounds.
T = thickness in inches, not less than 5/16 in.
D = mean diameter in inches.
C = 14,000, a constant, when corrugations are not more than 8 in. from center to center and not less than 1 1/2 in. deep.

PURVES TYPE

$$P=\frac{C\times T}{D}$$

Where P = pressure in pounds.
T = thickness in inches, not less than 7/16 in.
Where D = least outside diameter in inches.
C = 14,000, a constant, when rib projections are not more than 9 in. from center to center and not less than 1 3/8 in. deep.

BROWN TYPE

$$P = \frac{C \times T}{D}$$

Where P = pressure in pounds.

T = thickness in inches, not less than 5/16.

D = least outside diameter in inches.

C = 14,000, a constant (ascertained by an actual destruction test under the supervision of this Board), when corrugations are not more than 9 in. from center to center and not less than 1 5/8 in. deep.

The thickness of corrugated and ribbed furnaces shall be ascertained by actual measurement. The manufacturer shall have said furnace drilled for a 1/4-in. pipe tap and fitted with a screw plug that can be removed by the inspector when taking this measurement. For the Brown and Purves furnaces the holes shall be in the center of the second flat; for the Morison, Fox, and other similar types in the center of the top corrugation, at least as far in as the fourth corrugation from the end of the furnace.

TYPE HAVING SECTIONS 18 IN. LONG

$$P = \frac{C \times T}{D}$$

Where P = pressure in pounds.

T = thickness in inches, not less than 7/16.

D = mean diameter in inches.

$C = 10{,}000$, a constant, when corrugated by sections not more than 18 in. from center to center and not less than 2 1/2 in. deep, measuring from the least inside to the greatest outside diameter of the corrugations, and having the ends fitted one into the other and substantially riveted together, provided that the plain parts at the ends do not exceed 12 in. in length.

TOPS OF COMBUSTION CHAMBERS AND BACK CONNECTIONS

Formula for girders over back connection and other flat surfaces:

$$\text{Working pressure} = \frac{C \times d^2 \times T}{(W-P) \times D \times L}$$

Where W = width of combustion box in inches.

P = pitch of supporting bolts in inches.

D = distance between girders from center to center in inches.

L = length of girder in feet.

d = depth of girder in inches.

T = thickness of girder in inches.

$C = 550$ when the girder is fitted with one supporting bolt.

$C = 825$ when the girder is fitted with two or three supporting bolts.

$C = 917$ when the girder is fitted with four or five supporting bolts.

$C = 963$ when six or seven supporting bolts are used.

$C = 990$ when eight or more supporting bolts are used.

EXAMPLE

Given $W = 34$ in., $P = 7.5$ in., $D = 7.75$ in., $L = 2.927$ ft., $d = 7.5$ in., $T = 2$ in., $C = 825$, then, substituting in formula,

$$\text{Working pressure} = \frac{825 \times 7.5 \times 7.5 \times 2}{(34 - 7.5) \times 7.75 \times 2.927} = 154.3 \text{ lb.}$$

FLAT SURFACES

The maximum stress allowable on flat plates supported by stays shall be determined by the following formula:

All stayed surfaces formed to a curve the radius of which is over 21 in., excepting surfaces otherwise provided for, shall be deemed flat surfaces.

$$\text{Working pressure} = \frac{C \times T^2}{P^2}$$

Where T = thickness of plates in sixteenths of an inch.

P = greatest pitch of stays in inches.

$C = 112$ for screw stays with riveted heads, plates 7/16 in. thick and under.

$C = 120$ for screw stays with riveted heads, plates above 7/16 in. thick.

$C = 120$ for screw stays with nuts, plates 7/16 in. thick and under.

$C = 125$ for screw stays with nuts, plates above 7/16 in. thick and under 9/16 in.

$C = 135$ for screw stays with nuts, plates 9/16 in. thick and above.

$C = 175$ for stays with double nuts having one nut on the inside and one nut on the outside of plate, without washers or doubling plates.

$C = 160$ for stays fitted with washers or doubling strips which have a thickness of at least .5 of the thickness of the plate and a diameter of at least .5 of the greatest pitch of the stay, riveted to the outside of the plates, and stays having one nut inside of the plate, and one nut outside of the washer or doubling strip. For T take 72 per cent. of the combined thickness of the plate and washer or plate and doubling strip.

$C = 200$ for stays fitted with doubling plates which have a thickness equal to at least .5 of the thickness of the plate reenforced, and covering the full area braced (up to the curvature of the flange if any) riveted to either the inside or outside of the plate, and stays having one nut outside and one inside of the plates. Washers or doubling plates to be substantially riveted. For T take 72 per cent. of the combined thickness of the two plates.

$C = 200$ for stays with plates stiffened with tees or angle bars having a thickness of at least two-thirds the thickness of plate and depth of webs at least one-fourth of the greatest pitch of the stays, and substantially riveted on the inside of the plates,

and stays having one nut inside bearing on washers fitted to the edges of the webs that are at right angles to the plate. For T take 72 per cent. of the combined thickness of web and plate.

No such flat plates or surfaces shall be unsupported at a greater distance than 18 in.

REQUIREMENTS FOR HEADS

All plates used as heads, when new and made to practically true circles, and as described below, shall be allowed a steam pressure in accordance with the following formula:

CONVEX HEADS

$$P = \frac{T \times S}{R}$$

Where P = steam pressure allowable in pounds.
T = thickness of plate in inches.
S = one-fifth of the tensile strength.
R = one-half of the radius to which the head is bumped.

CONCAVE HEADS

For concave heads the pressure allowable will be .8 times the pressure allowable for convex heads.

NOTE.—To find the radius of a sphere of which the bumped head forms a part, square the radius of head, divide this by the height of bump required; to the result add height of bump, which will equal diameter of sphere, one-half of which will be the required radius.

Example.—Required the working pressure of a convex head of a 54-in. radius, material 60,000 lb. tensile strength and 1/2 in. thick.

Substituting values,

$$P = \frac{.5 \times 12000}{27} = 222 \text{ lb.}$$

The pressure allowable on a concave head of the same dimensions would be

$$222 \times .8 = 177 \text{ lb.}$$

ANGLE STIFFENERS FOR CURVED SURFACES

Where rounded bottoms of combustion chambers are stiffened with single angle-iron stiffeners, such angles shall have a thickness of leaf eight-tenths that of the plate and a depth of at least one-half pitch. Where stiffened with double angle irons or tee bars, such angles or tee bars shall have a thickness of leaf at least two-thirds that of plate and a depth of at least one-fourth of pitch. Said angles or tee bars shall be substantially riveted to the plate supported.

Where rounded tops of combustion chambers are stiffened with single or double angle-iron stiffeners, or tee bars, such angles or tee bars shall be of thickness and depth of leaf not less than specified for rounded bottoms of combustion chambers. Said angles or tee bars shall be supported on thimbles and riveted through with rivets not less than 1 in. in diameter, and spaced not to exceed 6 in. between centers.

Working pressure allowed on rounded surfaces supported by angle irons or tee bars shall be determined by the following formula:

$$\text{Working pressure} = \frac{C \times T^2}{P \times D}$$

Where T = thickness of plate in sixteenths of an inch.
P = pitch of angle or tee stiffeners in inches.
D = diameter of curve to which plate is bent, in inches.
C = 900, a constant.

Example.—Given $T=9/16$ in. $P=7$ in. $D=51$ in.
Substituting values in formula and solving,

$$\text{Working pressure} = \frac{900 \times 81}{7 \times 51} = 204 \text{ lb. per square inch}$$

PRESSURE PERMISSIBLE ON ROUNDED BOTTOM OF COMBUSTION CHAMBERS, ANGLES BEING OMITTED

$$P = \frac{50(300T - 2L)}{D}$$

Where P = working pressure in pounds.
T = thickness of bottom plate of combustion chamber in inches.
L = extreme length of plate forming bottom of combustion chamber in inches.
D = twice outside radius of bottom of combustion chamber in inches.

Example.—Required the working pressure on the bottom plate of a combustion chamber, angles being omitted: Thickness of plate, .82 in., extreme length of plate, 33 in., twice the radius of bottom of combustion chamber, 50 in. Substituting:

$$P = \frac{50 \times (300 \times .82 - 2 \times 33)}{50} = 180 \text{ lb.}$$

$$T = \frac{P \times D + 100L}{15000}$$

Pressure allowable on tube sheets where combustion chambers are not suspended from the shell of the boiler shall be determined by the following formula:

$$P = \frac{(D-d) \times T \times 27000}{W \times D}$$

Where P = working pressure in pounds.
D = least horizontal distance between tube centers, in inches.
d = inside diameter of tubes in inches.
T = thickness of tube plates in inches.
W = extreme width of combustion chamber in inches.

The compressive stress on tube plates, as determined by the following formula, must not exceed 13,500 lb. per square inch, when pressure on top of combustion chamber is supported by vertical plates of such chamber.

$$C = \frac{P \times D \times W}{2(D-d)\,T}$$

Where C = stress on tube sheet.
P = working pressure in pounds.
D = least horizontal distance between tube centers in inches.
d = inside diameter of tubes in inches.
W = extreme width of combustion chamber in inches.
T = thickness of tube sheet in inches.

Safety Valves

The areas of safety valves shall be determined in accordance with the following formula and table:

$$a = .2074 \times \frac{W}{P}$$

Where a = area of safety valve, in square inches, per square foot of grate surface.

W = pounds of water evaporated per square foot of grate surface per hour.

P = absolute pressure per square inch = working gage pressure + 15.

From which formula the areas required per square foot of grate surface in the following table are found by assuming the different values of W and P.

The figures (a) in table multiplied by square feet of grate surface give the area of safety valve or valves required.

When this calculation results in an odd size of safety valve, use next larger standard size.

Examples.—Boiler pressure = 75 lb. per square inch (gage).

2 furnaces: Grate surface = 2 (No.) × 5 ft. 6 in. (long) × 3 ft. (wide) = 33 sq. ft.

Water evaporated per pound of coal = 8 lb.

Coal burned per square foot grate surface per hour = 12 1/2 lb.

Evaporation per square foot grate surface per hour = 8 × 12 1/2 = 100 lb.

Hence W = 100 and gage pressure = 75 lb.

From table the corresponding value of a is .230 sq. in.

Therefore area of safety valve = 33 × .23 = 7.59 sq. in.

For which the diameter is 3 1/8 in. nearly.

Boiler pressure = 215 lb.

TABLE OF AREA OF SAFETY VALVES REQUIRED PER SQUARE FOOT OF GRATE SURFACE FOR DIFFERENT PRESSURES AND RATES OF EVAPORATION

P, absolute pressure per square inch	Gage pressure per square inch	These figures represent evaporation in pounds per square foot of grate surface per hour (W) = pounds water evaporated per pound coal × pounds coal burned per square foot of grate surface per hour														
		100	120	140	160	180	200	220	240	260	280	300	320	340	360	380
		The figures below give a, the area in square inches required per square foot of grate surface at the above rate of evaporation														
65	50	.319	.383	.447	.510	.574	.638	.702	.765	.829	.893	.956	.....	.....	.....	.....
70	55	.296	.355	.414	.474	.533	.592	.652	.711	.769	.828	.888	.....	.....	.....	.....
75	60	.276	.332	.387	.442	.497	.552	.608	.663	.718	.773	.829	.....	.....	.....	.....
80	65	.259	.311	.363	.415	.466	.518	.570	.622	.674	.726	.778	.....	.....	.....	.....
85	70	.244	.292	.341	.390	.438	.487	.536	.585	.634	.682	.731	.....	.....	.....	.....
90	75	.230	.276	.322	.368	.414	.460	.506	.552	.598	.644	.690	.....	.....	.....	.....
95	80	.218	.262	.305	.349	.392	.436	.479	.523	.567	.610	.654	.....	.....	.....	.....
100	85	.207	.249	.290	.332	.373	.414	.456	.497	.538	.580	.622	.....	.....	.....	.....
105	90	.197	.236	.276	.316	.355	.394	.434	.473	.513	.552	.592	.....	.....	.....	.....
110	95	.188	.226	.264	.301	.339	.377	.414	.452	.489	.527	.565	.....	.....	.....	.....
115	100	.180	.216	.252	.288	.324	.360	.396	.432	.468	.504	.540	.....	.....	.....	.....
120	105	.172	.207	.241	.276	.311	.345	.379	.414	.448	.483	.517	.....	.....	.....	.....
125	110	.166	.199	.232	.265	.298	.331	.364	.397	.431	.463	.497	.....	.....	.....	.....
130	115	.160	.192	.223	.255	.287	.319	.351	.383	.415	.447	.479	.....	.....	.....	.....
135	120	.153	.184	.215	.246	.276	.307	.337	.368	.398	.429	.460	.....	.....	.....	.....
140	125	.148	.177	.207	.237	.266	.296	.325	.355	.385	.414	.444	.....	.....	.....	.....
145	130	.143	.172	.201	.229	.258	.287	.315	.344	.372	.401	.430	.....	.....	.....	.....
150	135	.138	.166	.194	.222	.249	.277	.304	.332	.360	.387	.415	.....	.....	.....	.....
155	140	.134	.160	.187	.214	.241	.268	.294	.321	.348	.375	.401	.....	.....	.....	.....
160	145	.130	.156	.181	.207	.233	.259	.285	.311	.337	.363	.389	.....	.....	.....	.....
165	150	.126	.151	.176	.201	.226	.251	.276	.301	.326	.352	.378	.....	.....	.....	.....
170	155	.122	.146	.171	.195	.219	.244	.268	.292	.317	.341	.366	.....	.....	.....	.....
175	160	.118	.142	.166	.189	.213	.236	.260	.284	.308	.331	.355	.....	.....	.....	.....
180	165	.115	.138	.161	.184	.207	.230	.254	.277	.300	.323	.346	.....	.....	.....	.....
185	170	.112	.135	.157	.179	.202	.224	.247	.269	.291	.314	.336	.....	.....	.....	.....
190	175	.109	.131	.153	.175	.196	.218	.240	.262	.284	.306	.328	.....	.....	.....	.....
195	180	.106	.128	.149	.170	.191	.213	.234	.255	.277	.298	.319	.....	.....	.....	.....

TABLE OF AREA OF SAFETY VALVES REQUIRED PER SQUARE FOOT OF GRATE SURFACE FOR DIFFERENT PRESSURES AND RATES OF EVAPORATION (*Continued*)

P, absolute pressure per square inch	Gage pressure per square inch	These figures represent evaporation in pounds per square foot of grate surface per hour (W) = pounds water evaporated per pound coal × pounds coal burned per square foot of grate surface per hour														
		100	120	140	160	180	200	220	240	260	280	300	320	340	360	380
		The figures below give a, the area in square inches required per square foot of grate surface at the above rate of evaporation														
200	185	.104	.124	.145	.166	.187	.207	.228	.249	.270	.290	.310	.....	.....	.....	.....
205	190	.101	.121	.142	.162	.182	.202	.223	.243	.263	.283	.303	.....	.....	.....	.....
210	195	.099	.119	.138	.158	.178	.198	.217	.237	.257	.277	.297	.....	.....	.....	.....
215	200	.096	.116	.135	.154	.173	.193	.212	.231	.250	.269	.289	.308	.327	.347	.366
220	205	.094	.113	.132	.151	.170	.189	.208	.226	.245	.264	.283	.302	.321	.340	.358
225	210	.092	.110	.129	.147	.166	.184	.203	.221	.240	.258	.276	.295	.314	.332	.350
230	215	.090	.108	.126	.144	.162	.180	.198	.216	.235	.253	.270	.289	.307	.325	.343
235	220	.088	.106	.124	.141	.159	.176	.194	.212	.229	.247	.264	.282	.300	.318	.326
240	225	.086	.104	.121	.138	.155	.173	.190	.207	.225	.242	.259	.276	.294	.311	.329
245	230	.085	.102	.119	.135	.152	.170	.186	.203	.220	.237	.254	.271	.288	.305	.322
250	235	.083	.100	.117	.133	.149	.167	.183	.199	.216	.233	.249	.266	.282	.299	.315
255	240	.081	.098	.114	.130	.146	.163	.179	.195	.211	.228	.244	.261	.277	.293	.309
260	245	.080	.096	.112	.128	.144	.160	.176	.192	.208	.224	.240	.255	.271	.287	.303
265	250	.078	.094	.110	.125	.141	.157	.172	.188	.203	.219	.235	.250	.266	.282	.298
270	255	.077	.092	.107	.123	.138	.153	.169	.184	.199	.215	.230	.245	.261	.276	.291
275	260	.075	.090	.105	.121	.136	.151	.166	.181	.196	.211	.226	.241	.256	.271	.286
280	265	.074	.089	.104	.118	.133	.148	.163	.178	.192	.207	.222	.237	.251	.266	.281
285	270	.073	.087	.102	.116	.131	.146	.160	.175	.189	.204	.218	.233	.247	.262	.276
290	275	.072	.086	.100	.114	.129	.143	.157	.172	.186	.200	.214	.228	.242	.257	.271
295	280	.070	.084	.098	.112	.127	.141	.154	.169	.182	.196	.210	.224	.238	.253	.267
300	285	.069	.083	.096	.110	.124	.138	.151	.166	.179	.193	.207	.221	.235	.249	.263
305	290	.068	.082	.095	.109	.122	.136	.149	.163	.177	.190	.204	.217	.231	.245	.258
310	295	.067	.080	.093	.107	.120	.134	.147	.160	.174	.187	.201	.214	.227	.241	.254
315	300	.066	.079	.092	.105	.118	.132	.145	.158	.171	.184	.197	.210	.223	.237	.250

6 furnaces: Grate surface = 6 (No.) × 5 ft. 6 in. (long) × 3 ft. 4 in. (wide) = 110 sq. ft.

Water evaporated per pound coal = 10 lb.

Coal burned per square foot grate surface per hour = 30 lb.

Evaporation per square foot grate surface per hour = 10 × 30 = 300 lb.

Hence $W = 300$, gage pressure = 215, and $a = .270$ (from table).

Therefore area of safety valve = 110 × .270 = 29.7 sq. in., which is too large for one valve. Use two.

$$\frac{29.7}{2} = 14.85 \text{ sq. in.} \quad \text{Diameter} = 4\ 3/8 \text{ in.}$$

To determine the area of a safety valve for boiler using oil as fuel or for boilers designed for any evaporation per hour,

Divide the total number of pounds of water evaporated per hour by any number of pounds of water evaporated per square foot of grate surface per hour (W) taken from, and within the limits of, the table. This will give the equivalent number of square feet of grate surface for boiler; for estimating the area of valve, then apply the table as in previous examples.

Example.—Required the area of a safety valve for a boiler using oil as fuel, designed to evaporate 8000 lb. of water per hour, at 175-lb. gage pressure.

Make $W = 200$.

$$\frac{8000}{200} = 40, \text{ the equivalent}$$

grate surface in square feet.

For gage pressure = 175 lb. and $W = 200$, from table, $a = .218$ sq. in.; .218 × 40 = 8.72 sq. in., the total area required for this boiler, for which the diameter is 3 5/16 in. closely.

WATER TUBE AND COIL BOILERS

The working pressure allowable on cylindrical shells of water tube or coil boilers, when such shells have a row

or rows of pipes or tubes inserted therein, shall be determined by the following formula:

$$P = \frac{(D-d) \times T \times S}{D \times R}$$

Where P = working pressure allowable in pounds.

D = distance in inches between the tube or pipe centers in a line from head to head.

d = diameter of hole in inches.

T = thickness of plate in inches.

S = one-sixth of the tensile strength of the plate.

R = radius of shell in inches.

n = number of tube holes in a pitch. When tubes on any one row are pitched unequally, nd must be substituted in the formula for d; where rows of tubes are pitched diagonally, each diagonal ligament shall not be less than three-fifths of each longitudinal ligament.

Example.—Required the working pressure of a cylindrical shell having holes 1 in. in diameter, spaced 2 in. from center to center, in a line from head to head; material, 1/2 in. thick; diameter of shell, 20 in.; tensile strength of plate, 60,000 lb.

Substituting values, we have

$$P = \frac{(2-1) \times .5 \times 10000}{2 \times 10} = 250 \text{ lb.}$$

PORCUPINE-TYPE BOILERS

The formula for determining pressure on boilers of the so-called Porcupine and similar types shall be as follows:

Multiply the vertical distance between the centers of the horizontal rows of tubes in inches by one-half the diameter of shell of boiler in inches, which gives the area

upon which the pressure is exerted to break a diagonal ligament, then find the sectional area of the ligament at its smallest part and multiply by one-sixth the tensile strength of the material. This result, divided by the area upon which the strain is exerted, gives the working pressure per square inch, which is as follows $\frac{EFT}{CD}=W$, the working pressure, in which E equals width of ligament in inches, F thickness of material in inches, T one-sixth of the tensile strength, C distance between vertical centers, and D one-half the inside diameter of the shell or central column.

For the boiler proposed, 30 in. diameter, 5/8 in. thick, tensile strength 60,000 lb., 1.219 in. would be width of ligament, .625 thickness of plate, 10,000 one-sixth of tensile strength, 3 11/16=3.6875 in., distance of vertical centers; 15 in., one-half the diameter of shell, would be as follows: 1.219 multiplied by .625, this product multiplied by one-sixth the tensile strength, 10,000, equals 7618.75. This product, divided by the product of 3.6875, distance between vertical centers, multiplied by 15, one-half the diameter, equals 55.3125, gives 137.7 as pressure allowed.

EXTRACTS FROM BOARD OF BOILER RULES, STATE OF MASSACHUSETTS

Maximum Pressure on Boilers

1. The maximum pressure allowed on any steam boiler constructed wholly of cast-iron shall not be greater than twenty-five (25) pounds to the square inch.

2. The maximum pressures allowed on any steam boiler, the tubes of which are secured to cast-iron headers, shall not be greater than one hundred and sixty (160) pounds to the square inch.

3. The maximum pressure allowed on any steam boiler constructed of iron or steel shells or drums shall be calculated from the inside diameter of the outside course, the percentage of strength of the longitudinal joint and the minimum thickness of the shell plates; the tensile strength of shell plates to be taken as fifty-five thousand (55,000) pounds per square inch for steel and forty-five thousand (45,000) pounds per square inch for iron when the tensile strength is not known.

SHEARING STRENGTH OF RIVETS

4. The maximum shearing strength of rivets per square inch of cross-section of area to be taken as follows:

	Pounds
Iron rivets in single shear	38,000
Iron rivets in double shear	70,000
Steel rivets in single shear	42,000
Steel rivets in double shear	78,000

FACTORS OF SAFETY

5. The lowest factors of safety used for steam boilers, the shells or drums of which are directly exposed to the products of combustion and the longitudinal joints of which are of lap-riveted construction, shall be as follows:

(a) Five (5) for boilers not over ten years old.

(b) Five and five-tenths (5.5) for boilers over ten and not over fifteen years old.

(c) Five and seventy-five hundredths (5.75) for boilers over fifteen and not over twenty years old.

(d) Six (6) for boilers over twenty years old.

(e) Five (5) on steam boilers, the longitudinal joints of which are of lap-riveted construction, and the shells of drums of which are not directly exposed to the products of combustion.

(f) Four and five-tenths (4.5) on steam boilers, the longitudinal joints of which are of butt and strap construction.

Fusible Plugs

1. Fusible plugs as required by Section 20, Chapter 465, Acts of 1907, shall be filled with pure tin.

2. The least diameter of fusible metal shall not be less than one-half inch, except for working pressure of over one hundred and seventy-five (175) pounds gage, or when it is necessary to place a fusible plug in a tube, in which cases the least diameter of fusible metal shall not be less than three-eighth (3/8) inch.

3. The location of fusible plugs shall be as follows:

(a) In Horizontal Return Tubular Boilers—in the back head, not less than two (2) inches above the upper row of tubes, and projecting through the sheet not less than one (1) inch.

(b) In Horizontal Flue Boilers—in the back head, on a line with the highest part of the boiler exposed to the production of combustion, and projecting through the sheet not less than one (1) inch.

(c) In Locomotive Type or Star Water Tube Boilers—

in the highest part of the crown sheet, and projecting through the sheet not less than one (1) inch.

(d) In Vertical Fire Tube Boilers—in an outside tube, placed not less than one-third (1/3) the length of the tube above the lower tube sheet.

(e) In Vertical Submerged Tube Boilers—in the upper tube sheet.

(f) In Water Tube Boilers, Horizontal Drums, Babcock & Wilcox Type—in the upper drum, not less than six (6) inches above the bottom of the drum, and over the first pass of the products of combustion, projecting through the sheet not less than one (1) inch.

(g) In Stirling Boilers, Standard Type—in the front side of the middle drum, not less than six (6) inches above the bottom of the drum, and projecting through the sheet not less than one (1) inch.

(h) In Stirling Boilers, Superheater Type—in the front drum, not less than six (6) inches above the bottom of the drum, and exposed to the products of combustion, projecting through the sheet not less than one (1) inch.

(i) In Water Tube Boilers, Heine Type—in the front course of the drum, not less than six (6) inches above the bottom of the drum, and projecting through the sheet not less than one (1) inch.

(j) In Robb-Mumford Boilers, Standard Type—in the bottom of the steam and water drum, twenty-four (24) inches from the center of the rear neck, and projecting through the sheet not less than one (1) inch.

(k) In Water Tube Boilers, Almy Type—in the tube directly exposed to the products of combustion.

(l) In Vertical Boilers, Climax or Hazelton Type—in a tube or center drum not less than one-half (1/2) the height of the shell, measuring from the lowest circumferential seam.

(m) In Cahall Vertical Water Tube Boilers—in the inner sheet of the top drum, not less than six (6) inches above the upper tube sheet.

(n) In Scotch Marine Type Boilers—in combustion chamber top, and projecting through the sheet not less than one (1) inch.

(o) In Dry Back Scotch Type Boilers—in rear head, not less than two (2) inches above the top row of tubes, and projecting through the sheet not less than one (1) inch.

(p) In Economic Type Boilers—in the rear head above the upper row of tubes.

(q) In Cast-iron Sectional Heating Boilers—in a section over and in direct contact with the products of combustion in the primary combustion chamber.

(r) For other types and new designs, fusible plugs shall be placed at the lowest permissible water level, in the direct path of the products of combustion, as near the primary combustion chamber as possible.

Size of Rivets

1. When the size of the rivets in the longitudinal joints of a boiler is not known, the diameter and cross-sectional area of rivet, after driving, shall be taken as follows:

Thickness of plate.	7/16 in.	7/16 in.	15/32 in.	1/2 in.	9/16 in.	5/8 in.
Diameter of rivet after driving.	7/8 in. up to 2 1/4 in. pitch	15/16 in. over 2 1/4 in. pitch	15/16 in.	15/16 in.	1 1/16 in.	1 1/16 in.
Cross-sectional area of rivet after driving.	.6013 sq. in.	.6903 sq. in.	.6903 sq. in.	.6903 sq. in.	.8866 sq. in.	.8866 sq. in.

Thickness of plate.	1/4 in.	9/32 in.	5/16 in.	11/32 in.	3/8 in.	3/8 in.	13/32 in.
Diameter of rivet after driving.	1 1/16 in.	1 1/16 in.	3/4 in.	3/4 in.	3/4 in. up to 2 in. pitch	13/16 in. over 2 in. pitch	13/16 in.
Cross-sectional area of rivet after driving.	.3712 sq. in.	.3712 sq. in.	.4418 sq. in.	.4418 sq. in.	.4418 sq. in.	.5185 sq. in.	.5185 sq. in.

Allowable Strain on Stays

1. The maximum allowable strain per square inch net cross-section for weldless mild steel shall be as follows:

Type	Size up to and including 1 1/2 in. diameter or equivalent	Size over 1 1/2 in. diameter or equivalent
Head to head or through stays.	8,000 lb.	9,000 lb.
Diagonal or crowfoot stays....	7,500 lb.	8,000 lb.
Screwed stays (stay bolts).....	7,000 lb.	7,000 lb.

2. For welded stays the strain allowed per square inch net cross-section shall not exceed six thousand (6000) pounds.

3. For wrought-iron stays or stay bolts the strain allowed per square inch net cross-section shall not exceed six thousand (6000) pounds.

		$W=75/3600$ $P=40$ $A=.401$	$W=100/3600$ $P=65$ $A=.329$	$W=160/3600$ $P=115$ $A=.297$	$W=160/3600$ $P=140$ $A=.244$	$W=200/3600$ $P=190$ $A=.224$	$W=240/3600$ $P=240$ $A=.213$
Gage pressure per square inch at which safety valve is set to blow		Zero to 25 lb.	Over 25 to 50 lb.	Over 50 to 100 lb.	Over 100 to 150 lb.	Over 150 to 200 lb.	Over 200 lb.
Diameter of valve in inches	Area of valve in square inches	Area of grate in square feet					
1	.7854	2.0	2.4	2.7	3.2	3.5	3.7
1 1/4	1.2272	3.1	3.8	4.2	5.0	5.5	5.7
1 1/2	1.7671	4.5	5.4	6.0	7.2	7.9	8.3
2	3.1416	7.9	9.6	10.6	12.9	14.0	14.8
2 1/2	4.9087	12.3	15.0	16.5	20.0	21.9	23.0
3	7.0686	17.6	21.3	23.8	29.0	31.5	33.2
3 1/2	9.6211	24.0	29.3	32.4	39.4	42.9	45.2
4	12.5660	31.4	38.2	42.3	51.5	56.0	59.0
4 1/2	15.9040	40.0	48.4	53.5	65.0	71.0	74.7
5	19.6350	49.0	60.0	66.0	80.0	88.0	92.1

Appendages to be Placed on Boilers

1. Each boiler shall have a safety valve the minimum area of which shall be in accordance with the following tables. If more than one safety valve is used the minimum combined area shall be in accordance with the following tables.

2. When the conditions exceed those on which the tables are based the formula shall be used.

3. A table of areas of grate surface in square feet for pop safety valves follows:

$$A = \frac{W70}{P} \times 11$$

A = Area of safety valve in square inches per square foot of grate.
W = Weight of steam per second.
P = Pressure, absolute.

4. A table of grate areas in square feet for safety valves (other than pop safety valves) follows; this table is in ratio to the table for pop safety valves as 2 is to 3:

Gage pressure per square inch at which safety valve is set to blow		Zero to 25 lb.	Over 25 to 50 lb.	Over 50 to 100 lb.
Diameter of valve in inches	Area of valve in square inches	Area of grate in square feet		
1	.7854	1.4	1.6	1.8
1 1/4	1.2272	2.1	2.5	2.8
1 1/2	1.7671	3.0	3.6	4.0
2	3.1416	5.3	6.4	7.1
2 1/2	4.9087	8.2	10.0	11.0
3	7.0686	11.7	14.2	16.0
3 1/2	9.6211	16.0	19.5	21.6
4	12.5660	21.0	25.5	28.2
4 1/2	15.9040	26.7	32.3	36.0
5	19.6350	32.7	40.0	44.0

5. Each safety valve must have full-sized direct connection to the boiler, and full-sized escape pipe which shall be fitted with an open drain to prevent water lodging in the upper part of safety valve or escape pipe. When a boiler is fitted with two safety valves on one connection this connection to the boiler shall have a cross-sectional area equal to or greater than the combined area of the two safety valves.

6. Safety valves having either the seat or disc of cast-iron shall not be used.

7. The seats of all safety valves shall be inclined at an angle of forty-five (45) degrees to the center line of the spindle.

8. A certificate of inspection shall not be issued on a boiler used for heating purposes exclusively, permitting the boiler to be operated at a pressure in excess of fifteen (15) pounds, if the boiler is provided with a device (safety valve) in accordance with the provision contained in section 78, chapter 102 of the Revised Laws, limiting the pressure carried to fifteen (15) pounds.

9. Each boiler shall have a steam gage connected to the steam space of the boiler by a syphon, or equivalent device, sufficiently large to fill the gage tube with water, and in such manner that the steam gage cannot be shut off from the boiler except by a cock with T end, placed directly on the pipe under the steam gage.

10. The dial of the steam gage shall be graduated to not less than one and one-half (1 1/2) times the maximum pressure allowed on the boiler.

11. Each boiler shall be provided with a one-eighth (1/8) inch pipe size connection for attaching inspector's

test gage when boiler is in service, so that the accuracy of the boiler steam gage can be ascertained, as required by section 3, chapter 465, Acts of 1907.

12. Each boiler shall have one fusible plug, as required by rules (section 3) on fusible plugs.

13. Each boiler shall have one water-glass, the bottom end of which shall be above the fusible plug and lowest safe water line.

14. Each boiler shall have two or more gage cocks, located within the range of the water-glass, when the maximum pressure allowed does not exceed twenty-five (25) pounds per square inch.

15. Each boiler shall have three or more gage cocks, located within the range of the water-glass, when the maximum pressure allowed exceeds twenty-five (25) pounds per square inch.

16. Each steam outlet from boiler shall be fitted with a stop valve.

17. When a stop valve is so located that water can accumulate, ample drains shall be provided.

18. Each boiler shall have a feed pipe fitted with check valve, and also a stop valve between the check valve and the boiler, the feed water to discharge below the lowest safe water line. Means must be provided for feeding the boiler with water when the maximum pressure allowed is carried on the boiler.

19. Each boiler shall have a bottom blow-off pipe fitted with a stop valve or stop cock, and connected direct to the lowest water space of the boiler.

20. Where a damper regulator is used, the boiler pressure pipe shall be taken from the steam space of the

boiler, and shall be fitted with a stop valve or stop cock.

21. Each boiler fitted with a Lamphrey Boiler Furnace Mouth Protector, or similar appendage, having valves on the pipes connecting same with the boiler, shall have these valves locked or sealed open, so that the locks or seals will require to be removed or broken to shut the valves.

Annual Internal Inspections

1. The owner or user of a steam boiler which requires annual inspection, internally and externally, by the boiler inspection department or by an insurance company, as provided by section 1, chapter 465, Acts of 1907, shall prepare the boiler for inspection by cooling it down (blanking off connections to adjacent boilers if necessary), removing all soot and ashes from tubes, heads, shell, furnace and combustion chamber; drawing off the water; removing the handhole and manhole plates; removing the grate bars from internally fired boilers; and removing the steam gage for testing.

2. If a boiler has not been properly cooled down, or otherwise prepared for inspection, the boiler inspector shall decline to inspect it, and he shall not issue a certificate of inspection until efficient inspection has been made.

3. In making the annual internal and external inspection as provided by sections 1 and 4, chapter 465, Acts of 1907, the boiler inspector shall apply the hammer test to all internal and external parts of a boiler that are accessible.

4. All proper measurements shall be taken by the boiler inspector, so that the maximum working pressure allowed on a boiler will conform to the rules relating to allowable pressures established by the Board of Boiler Rules; such measurements to be taken and calculations made before a hydrostatic pressure test is applied to a boiler.

5. The steam gage of a boiler shall be tested and its readings compared with an accurate test gage, and if, in the judgment of the boiler inspector, the gage is not reliable he shall order it repaired or replaced.

Annual External Inspections

1. The annual external inspection of a steam boiler, as provided for in section 3, chapter 465, Acts of 1907, should be made at or about six (6) months after the annual internal inspection, except in the case of a boiler that is in service a portion of the year only, in which case the annual external inspection shall be made during such period of service.

2. The boiler inspector shall attach an accurate test gage to a boiler to note the pressure shown by said test gage, and compare it with that shown by the boiler gage, ordering the boiler gage repaired or replaced if necessary.

3. The boiler inspector shall see that the water-glass, gage cocks, water-column connections and water blow-offs are free and clear; also that the safety valve raises freely from its seat.

4. Fire doors, tube doors, and doors in settings shall be opened, to view as far as possible the fire surface,

settings, tube ends, blow-off pipes and fusible plug, noting conditions and ordering changes or repairs if necessary.

Hydrostatic Pressure Tests

1. When a boiler is tested by hydrostatic pressure, the maximum pressure applied shall not exceed one and one-half (1 1/2) times the maximum working pressure allowed, except that twice the maximum working pressure allowed may be applied on boilers permitted to carry twenty-five (25) pounds pressure per square inch or less, or on pipe boilers.

2. When making annual inspections on boilers constructed wholly of cast iron, or on pipe boilers, a hydrostatic pressure test of not less than one and one-half (1 1/2) times and not more than twice the maximum working pressure allowed shall be applied.

3. The boiler inspector, after applying a hydrostatic pressure test, shall thoroughly examine every accessible part of the boiler, both internal and external.

TO DETERMINE MAXIMUM ALLOWABLE PRESSURE

Formula:

$\frac{T.S. \times t \times \%}{R \times F.S.}$ = maximum allowable working pressure, per square inch, in pounds.

$T.S.$ = Tensile strength of shell plates, in pounds.

t = minimum thickness of shell plates, in inches.

$\%$ = efficiency of longitudinal joint.

R = radius = one-half (1/2) the inside diameter of the outside course of the shell or drum.

$F.S.$ = lowest factor of safety allowed by these rules.

When the tensile strength of steel or wrought iron shell plates is *not* known, it shall be taken as fifty-five thousand (55,000) pounds for steel, and forty-five thousand (45,000) pounds for wrought iron.

Efficiency of Riveted Joints

The efficiency that a unit of length of a riveted joint has to the same unit of length of solid plate shall be calculated as shown by the following examples:

$T.S.$ = tensile strength of plate, in pounds per square inch.

t = thickness of plate, in inches.

b = thickness of butt strap, in inches.

P = pitch of rivets, in inches, on row having greatest pitch.

d = diameter of rivet after driving, in inches.

a = cross-sectional area of rivet after driving, in square inches.

s = strength of rivet in single shear.

S = strength of rivet in double shear.

C = crushing strength of mild steel.

(*Note.*—"C" applies only to boilers constructed after February 5, 1910.)

n = number of rivets in single shear in a unit of length of joint.

N = number of rivets in double shear in a unit of length of joint.

LAP, SINGLE-RIVETED.

LONGITUDINAL OR CIRCUMFERENTIAL

Example.

A = strength of solid plate $= P \times t \times T.S.$

B = strength of solid plate between rivet holes $= (P-d) \times t \times T.S.$

C = shearing strength of one rivet in single shear $= n \times s \times a.$

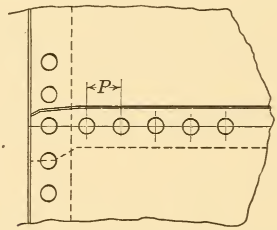


FIG. 1.—Single-riveted lap joint.

D = crushing strength of plate in front of one (1) rivet $= d \times t \times c.$

Divide B, C, or D (whichever is the least) by A, and the quotient will be the efficiency of a single-riveted lap joint.

$T.S. = 55{,}000$ lb.

$t = 1/4$ in. $= .25$ in.

$d = 11/16$ in. $= .6875$ in.

$a = .3712$ sq. in.

$s = 42{,}000$ lb.

$c = 95{,}000$ lb.

$A = 1.625 \times .25 \times 55{,}000 = 22{,}343.$

$B = (1.625 - .6875).25 \times 55{,}000 = 12.890.$
$C = 1 \times 42{,}000 \times .3712 = 15.590.$
$D = .6875 \times .25 \times 95{,}000 = 16{,}328.$

$$\frac{12890\ (B)}{22343\ (A)} = .576, \text{ efficiency of joint.}$$

(See Fig. 1 of this group.)

Lap, Double-riveted.

LONGITUDINAL OR CIRCUMFERENTIAL

Strength of solid plate $= P \times t \times T.S. = A.$

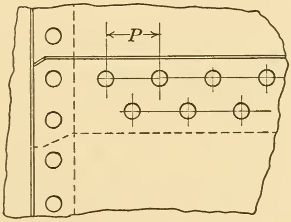


Fig. 2.—Double-riveted lap joint.

Strength of plate between rivet holes $= (P - d)t \times T.S. = B.$

Shearing strength of two (2) rivets in single shear $= nsa = C.$

Divide B or C (whichever is the least) by A, and the quotient will be the efficiency of a double-riveted lap joint.

$T.S. = 55{,}000$ lb.
$t = 5/16$ in. $= .3125$ in.
$P = 2\ 7/8$ in. $= 2.875$ in.

$d = 3/4$ in. $= .75$ in.
$a = .4418$ sq. in.
$s = 42,000$
$A = 2.875 \times .3125 \times 55,000 = 49,414.$
$B = (2.875 - .75) \times .3125 \times 55,000 = 36,523.$
$C = 2 \times .4418 \times 42,000 = 37,111.$

$\frac{36523\ (B)}{49414\ (A)} = .739$, effiiciency of joint.

(See Fig. 2 of this group.)

Butt, Double-riveted.

BUTT AND DOUBLE-STRAP JOINT

A = strength of solid plate $= P \times t \times T.S.$

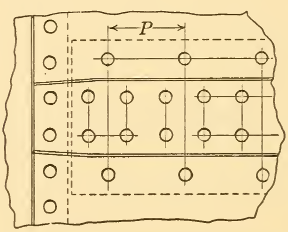


Fig. 3.—Double-riveted, double butt-strapped joint.

B = strength of plate between rivet holes in the outer row $= (P - d)t \times T.S.$

C = shearing strength of two (2) rivets in double shear; plus the shearing strength of one (1) rivet in single shear $= N \times S \times a + n \times s \times a.$

D = strength of plate between rivet holes in the second row, plus the shearing strength of one (1) rivet in single shear in the outer row = $(P-2d)t\times T.S.+n\times s\times a$.

E = strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one (1) rivet in the outer row = $(P-2d)t\times T.S.+d\times b\times c$.

F = crushing strength of plate in front of two (2) rivets, plus the crushing strength of butt strap in front of one (1) rivet = $N\times d\times t\times c+n\times d\times b\times c$.

G = crushing strength of plate in front of two (2) rivets, plus the shearing strength of one (1) rivet in single shear = $N\times d\times t\times c+n\times s\times a$.

Divide B, C, D, E, F, or G (whichever is the least) by A, and the quotient will be the efficiency of a butt and double-strap joint, double-riveted. (See Fig. 3 of this group.)

$T.S.=55{,}000$ lb.	$a=.6013$ sq. in.
$t=3/8$ in. $=.375$ in.	$s=42{,}000$ lb.
$b=5/16$ in. $=.3125$ in.	$S=78{,}000$ lb.
$P=4\ 7/8$ in. $=4.875$ in.	$c=95{,}000$ lb.

Number of rivets in single shear in a unit of length of joint = 1.

Number of rivets in double shear in a unit of length of joint = 2.

$$A=4.875\times.375\times55{,}000=100{,}547.$$

$$B=(4.875-.875).375\times55{,}000=82{,}500.$$

$C = 2 \times 78{,}000 \times .6013 + 1 \times 42{,}000 \times .6013 =$
$119{,}057.$

$D = (4.875 - 2 \times .875) .375 \times 55{,}000 + 1 \times 42{,}000 \times$
$.6013 = 89{,}708.$

$E = (4.875 - 2 \times .875) .375 \times 55{,}000 + .875 \times .3125$
$\times 95{,}000 = 90{,}429.$

$F = 2 \times .875 \times .375 \times 95{,}000 + .875 \times .3125 \times 95{,}000$
$= 88{,}320.$

$G = 2 \times .875 \times .375 \times 95{,}000 + 1 \times 42{,}000 \times .6013 =$
$87{,}599.$

$$\frac{82500(B)}{100547(A)} = .820, \text{ efficiency of joint.}$$

(See Fig. 3 in this group.)

BUTT, TRIPLE-RIVETED.

BUTT AND DOUBLE-STRAP JOINT

A = strength of solid plate $= P \times t \times T.S.$

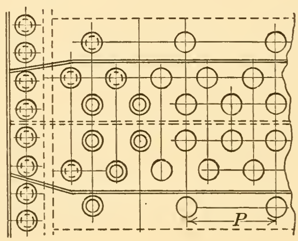


FIG. 4.—Triple-riveted, double butt-strapped joint.

B = strength of plate between rivet holes in the outer row $= (P - d) t \times T.S.$

C = shearing strength of four (4) rivets in double shear, plus the shearing strength of one (1) rivet in single shear $= N \times S \times a + n \times s \times a.$

D = strength of plate between the rivet holes in the second row, plus the shearing strength of one (1) rivet in single shear in the outer row $= (P-2d)t \times T.S. + n \times s \times a$.

E = strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one (1) rivet in the outer row $= (P-2d)t \times T.S. + d \times b \times c$.

F = crushing strength of plate in front of four (4) rivets, plus the crushing strength of butt strap in front of one (1) rivet $= N \times d \times t \times c + n \times d \times b \times c$.

G = crushing strength of plate in front of four (4) rivets, plus the shearing strength of one (1) rivet in single shear $= N \times d \times t \times c + n \times s \times a$.

Divide B, C, D, E, F, or G (whichever is the least) by A, and the quotient will be the efficiency of a butt and double-strap joint, triple riveted. (See Fig. 4 of this group.)

$T.S. = 55{,}000$ lb.	$a = .5185$ sq. in.
$t = 3/8$ in. $= .375$ in.	$s = 42{,}000$ lb.
$b = 5/16$ in. $= .3125$ in.	$S = 78{,}000$ lb.
$P = 6\ 1/2$ in. $= 6.5$ in.	$c = 95{,}000$ lb.
$d = 13/16$ in. $= .8125$	

Number of rivets in single shear in a unit of length of joint = 1.

Number of rivets in double shear in a unit of length of joint = 4.

$$A = 6.5 \times .375 \times 55{,}000 = 134{,}062.$$

$$B = (6.5 - .8125).375 \times 55{,}000 = 117{,}304.$$

$C=4\times 78{,}000\times .5185+1\times 42{,}000\times .5185=$ 183,549.

$D=(6.5-2\times .8125).375\times 55{,}000+1\times 42{,}000\times .5185=122{,}323.$

$E=(6.5-2\times .8125).375\times 55{,}000+.8125\times .3125\times 95{,}000=124{,}667.$

$F=4\times .8125\times .375\times 95{,}000+1\times .8125\times .3125\times 95{,}000=139{,}902.$

$G=4\times .8125\times .375\times 95{,}000+1\times 42{,}000\times .5185=137{,}558.$

$$\frac{117304(B)}{134062(A)}=.875, \text{ efficiency of joint.}$$

(See Fig. 4 in this group.)

Butt, Quadruple-riveted.

BUTT AND DOUBLE-STRAP JOINT, QUADRUPLE-RIVETED

A = Strength of solid plate $=P\times t\times T.S.$

B = Strength of plate between rivet holes in the outer row $=(P-d)t\times T.S.$

C = shearing strength of eight (8) rivets in double shear, plus the shearing strength of three (3) rivets in single shear $=N\times S\times a+n\times s\times a.$

D = strength of plate between rivet holes in the second row, plus the shearing strength of one (1) rivet in single shear in the outer row $=(P-2d)t\times T.S.+n\times s\times a.$

E = strength of plate between rivet holes in the third row, plus the shearing strength of two

(2) rivets in the second row in single shear and one (1) rivet in single shear in the outer row $=(P-4d)t \times T.S.+n \times s \times a.$

$F=$ strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one (1) rivet in the outer row $=(P-2d)t \times T.S.+d \times b \times c.$

$G=$ strength of plate between rivet holes in

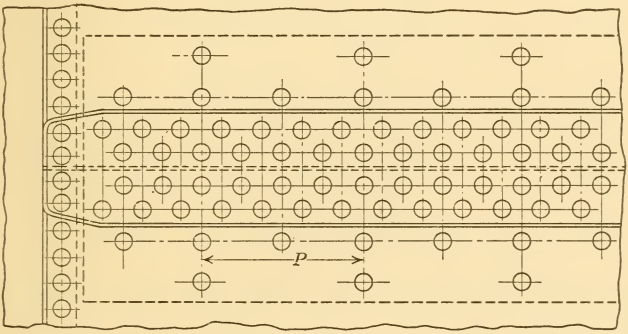


FIG. 5.—Quadruple-riveted, double butt-strapped joint.

the third row, plus the crushing strength of butt strap in front of two (2) rivets in the second row and one (1) rivet in the outer row $=(P-4d)t \times T.S.+n \times d \times b \times c.$

$H=$ crushing strength of plate in front of eight (8) rivets, plus the crushing strength of butt strap in front of three (3) rivets $=$ $N \times d \times t \times c+n \times d \times b \times c.$

$I=$ crushing strength of plate in front of eight (8) rivets, plus the shearing strength of two (2) rivets in the second row and one

(1) rivet in the outer row, in single shear $=$ $N \times d \times t \times c + n \times s \times a$.

Divide B, C, D, E, F, G, H, or I (whichever is the least) by A, and the quotient will be the efficiency of a butt and double-strap joint, quadruple-riveted.

$T.S. = 55,000$ lb.
$t = 1/2$ in. $= .5$ in.
$b = 7/16$ in. $= .4375$
$P = 15$ in.
$d = 15/16$ in. $= .9375$ in.
$a = .6903$ sq. in.
$s = 42,000$ lb.
$S = 78,000$ lb.
$c = 95,000$ lb.

Number of rivets in single shear in unit of length of joint $= 3$.

Number of rivets in double shear in unit of length of joint $= 8$.

$A = 15 \times .5 \times 55,000 = 412,500.$
$B = (15 - .9375) .5 \times 55,000 = 386,718.$
$C = 8 \times 78,000 \times .6903 + 3 \times 42,000 \times .6903 = 517,723.$
$D = (15 - 2 \times .9375) .5 \times 55,000 + 1 \times 42,000 \times .6903 = 389,930.$
$E = (15 - 4 \times .9375) .5 \times 55,000 + 3 \times 42,000 \times .6903 = 396,353.$
$F = (15 - 2 \times .9375) .5 \times 55,000 + .9375 \times .4375 \times 95,000 = 399,902.$

$G = (15 - 4 \times .9375).5 \times 55{,}000 + 3 \times .9375 \times .4375 \times 95{,}000 = 426{,}269.$

$H = 8 \times .9375 \times .5 \times 95{,}000 + 3 \times .9375 \times .4375 \times 95{,}000 = 473{,}145.$

$I = 8 \times .9375 \times .5 \times 95{,}000 + 3 \times 42{,}000 \times .6903 = 443{,}229.$

$\frac{386718(B)}{412500(A)} = .937$, efficiency of joint.

(See Fig. 5 of this group.)

Bumped Heads

The minimum thickness of a convex head shall be determined by the following formula:

$$\frac{R \times F.S. \times P}{T.S.} = t$$

The minimum thickness of a concave head shall be determined by the following formula:

$$\frac{R \times F.S. \times P}{.6(T.S.)} = t$$

R = one-half the radius to which the head is bumped.

$F.S.$ = 5 = factor of safety.

P = working pressure, in pounds per square inch, for which the boiler is designed.

$T.S.$ = tensile strength, in pounds per square inch, stamped on the head by the manufacturer.

t = thickness of the head in inches.

When a convex or a concave head has a manhole

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opening, the thickness as found by the preceding formulas shall be increased by not less than one-eighth (1/8) inch.

FORMULA TO FIND AREA OF SEGMENT OF CIRCLE TO BE BRACED

$$\frac{4 \times H^2}{3} \sqrt{\frac{2R}{H} - .608} = \text{area in square inches.}$$

H = distance from tubes to shell, minus five (5) inches.

R = radius of boiler, minus three (3) inches.

FORMULA FOR DIAMETER OF STAY BOLTS AT BOTTOM OF THREAD

$D - (P \times 1.732) = d$, or for 12 threads per inch,

$D - (.08333 \times 1.732) = d$, then

$D - .1443 = d$.

D = diameter of stay bolt over the threads.

P = pitch of threads $= 1/12 = .08333$.

d = diameter of stay bolt at bottom of threads.

1.732 = a constant.

When U. S. threads are used, the formula becomes: $D - (P \times 1.732 \times .75) = d$.

FORMULA FOR CAST-IRON NOZZLES

The minimum thickness of cast-iron nozzles shall be determined by the following formula:

$$\frac{Pdf}{2S} + .5 = t$$

P = working pressure in pounds per square inch.
d = inside diameter of nozzle in inches.
f = factor of safety = 12.
S = ultimate tensile strength of cast iron, not less than eighteen thousand (18,000) pounds per square inch.
.5 = a constant.
t = thickness of nozzle in inches.

Maximum Pressure on Boiler Shells

The maximum pressure to be allowed on a steel or wrought-iron shell or drum shall be determined from the minimum thickness of the shell plates, the lowest tensile strength stamped on the plates by the plate manufacturer, the efficiency of the longitudinal joint or ligament between the tube holes, whichever is the least, the inside diameter of the outside course, and a factor of safety of not less than five (5), the formula being:

$$\frac{T.S. \times t \times \%}{R \times F.S.} = \text{maximum allowable working pressure per square inch in pounds.}$$

$T.S.$ = tensile strength of shell plates in pounds.
t = minimum thickness of shell plates in inches.
$\%$ = efficiency of longitudinal joint or ligament between tube holes, whichever is the least.
R = radius = one-half (1/2) the inside diameter of the outside course of the shell or drum.
$F.S.$ = 5, the lowest factor of safety allowed on boilers installed after May 1, 1908.

The method of determining the efficiency of the longitudinal joint has already been explained and illustrated. To find the efficiency of ligaments, the following formulas are to be employed.

Efficiency of Ligaments

When a shell or drum is drilled for tube holes in a line parallel to the axis of the shell or drum, the efficiency of the ligament between the tube holes shall be determined as follows: (a) when the pitch of the tube holes on every row is equal the formula is:

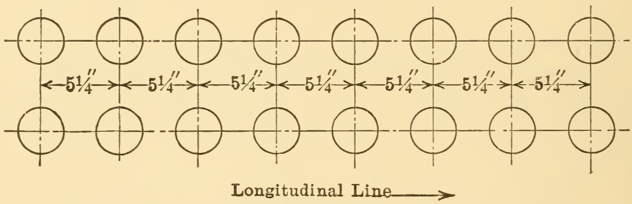

Fig. 1.—Diagram for calculating the efficiency of ligament.

$$\frac{p-d}{p} = \text{efficiency of ligament.}$$

p = pitch of tube holes in inches.
d = diameter of tube holes in inches.

Example.—Pitch of tube holes in the drum of a water-tube boiler = 5 1/4 in. = 5.25 in. Diameter of tube holes = 3 1/4 in. = 3.25 in.

$$\frac{p-d}{p} = \frac{5.25-3.25}{5.25} = .38, \text{ efficiency of ligament.}$$

(See Fig. 1 of this group.)

(b) when the pitch of the tube holes on any one row is unequal, the formula is:

$$\frac{P-nd}{P} = \text{efficiency of ligament.}$$

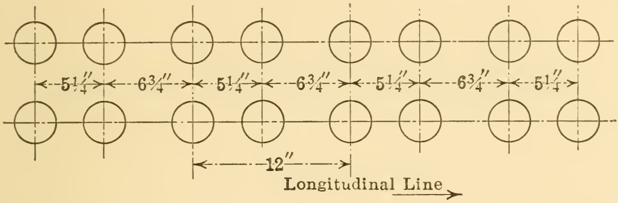

FIG. 2.—Diagram for calculating the efficiency of ligament.

P = unit length of ligament in inches.
n = number of tube holes in length, P.
d = diameter of tube holes in inches.

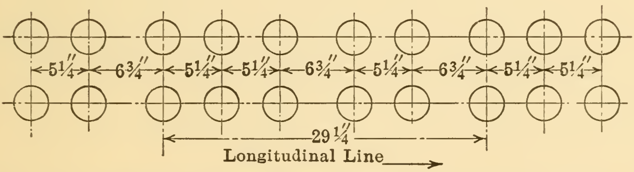

FIG. 3.—Diagram for calculating the efficiency of ligament.

Example.—

$$\frac{P-nd}{P} = \frac{12-2\times 3.25}{12} = .458, \text{ efficiency of ligament.}$$

(See Fig. 2 of this group.)

Example.—

$$\frac{P-nd}{P} = \frac{29.25-5\times 3.25}{29.25} = .444, \text{ efficiency of ligament.}$$

(See Fig. 3 of this group.)

When a shell or drum is drilled for tube holes in a line diagonal to the axis of the shell or drum, the efficiency of the ligament between the tube holes shall be determined as follows:

$$\frac{P-d}{p} = \text{efficiency of ligament.}$$

P = diagonal pitch of tube holes in inches.
d = diameter of tube holes in inches.
p = distance between rows of tubes longitudinally.

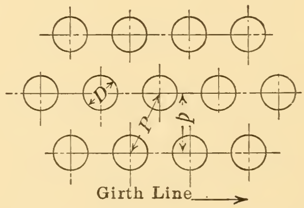

FIG. 4.—Diagram for calculating the efficiency of ligament.

Example.—Diagonal pitch of tube holes in a drum of a water-tube boiler = 6.42 in.
Diameter of tube holes = 4 in.
Distance between rows of tubes, longitudinally = 5.75 in.

$$\frac{P-d}{p} = \frac{6.42-4}{5.75} = .42, \text{ efficiency of ligament.}$$

(See Fig. 4 of this group.)

When a flat-head has a manhole opening, the flange of which is formed from the solid sheet and turned inward to a depth of not less than twice the thickness of the head, an area two (2) inches wide all around the manhole opening, as shown in the figure, may be deducted from the total area of head, including manhole opening, to be stayed.

Example.—To find an area 2 in. wide all around a 11 in. $\times$ 15 in. manhole,

$$15 \text{ in.} \times 19 \text{ in.} \times .7854 = 224 \text{ (nearly) sq. in.}$$
$$11 \text{ in.} \times 15 \text{ in.} \times .7854 = 130 \text{ (nearly) sq. in.}$$
$$\text{And } 224 - 130 = 94 \text{ sq. in.}$$

Therefore, if the area to be stayed on the *rear* head, below the tubes, of a seventy-two (72) inch horizontal return tubular boiler is 374 sq. in., the area to be stayed on *front* head, below the tubes, of this boiler, would be $374 - 94 = 280$ sq. in.

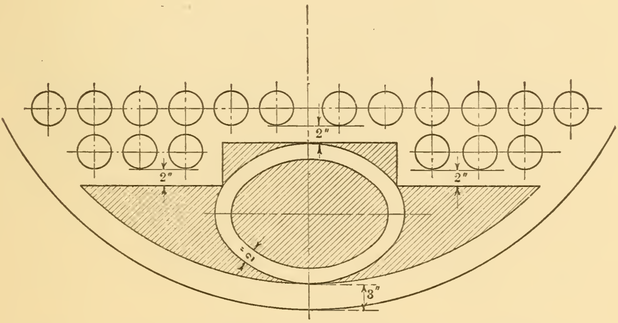

FIG. 1.—Diagram showing area of head to be braced.

A segment of a head of a horizontal return tubular, locomotive, Scotch or similar type shall be stayed by welded or weldless mild steel or wrought iron, head to head or through, diagonal or crow-foot stays, except a horizontal return tubular boiler, as otherwise provided for.

The area of a segment of a head to be stayed shall be the area enclosed by lines drawn three (3) inches from

the shell and two (2) inches from the tubes, as shown in Figs. 1 and 2.

When the shell of a horizontal return tubular boiler does not exceed thirty-six (36) inches in diameter, and is designed for a maximum working pressure not to exceed one hundred (100) pounds per square inch, the segment of the head above the tubes **may** be stayed by steel angles, or Tee bars, the formula being:

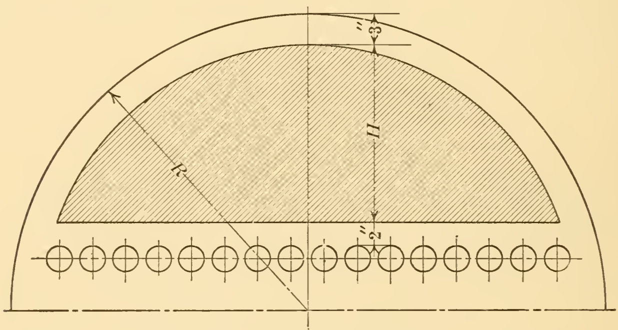

FIG. 2.—Diagram showing area of head to be braced.

$$\frac{fI}{y} = M$$

f = fiber stress = 16,000 lb.

I = moment of inertia $= \dfrac{bh^3}{12}$

h = height of beam in inches.
b = thickness of beam in inches.

y = distance of most strained fiber = h ÷ 2.

M = bending moment of beam.

maximum bending moment for a uniform load$=\frac{WL}{8}$.

$$\left\{\begin{array}{l} W = \text{weight to be supported in pounds.} \\ L = \text{length of beam in inches.} \end{array}\right\}$$

Example.—When steel angles are used, the head of a horizontal return tubular boiler thirty (30) inches in diameter, designed for

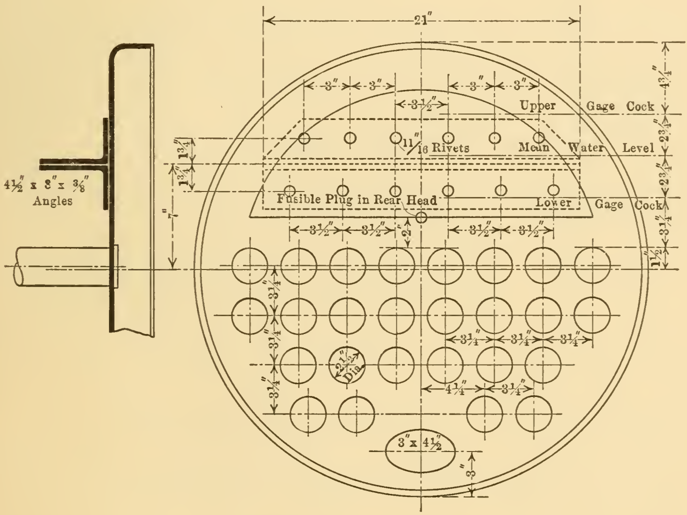

FIG. 1.—Diagram of steel angles bracing.

one hundred (100) pounds working pressure, shall be stayed by two (2) four and one-half by three by three-eighths (4 1/2×3×3/8) inch steel angles, as shown in the figure, or by other sized commercial steel angles, the resistance of which shall be equal to or greater than the maximum bending moment.

Distance from tubes to shell = 13 1/2 in.

Area to be stayed = 143 sq. in.
Load at 100 lb. pressure = 14.300 lb.

$$\frac{WL}{8} = \frac{14300 \times 21.}{8} = 37{,}540 \text{ lb.}$$

Moment of inertia $= I = 1/12 \times 4.5^3 \times 3/8 = 2.85.$
$y = 4.5 \div 2 = 2.25.$

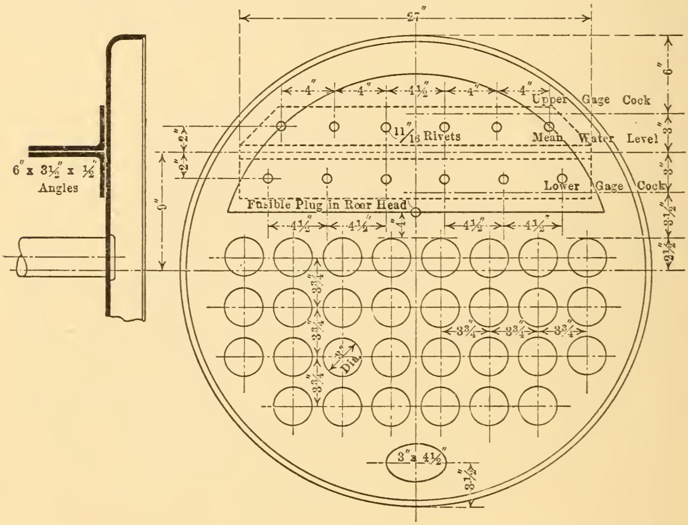

FIG. 2.—Diagram of steel angles bracing.

$$\frac{fI}{y} = M = \frac{16000 \times 2.85}{2.25} = 20{,}266 \text{ lb. for one angle.}$$

Resistance of one angle = 20,266 lb.
Resistance of two angles = 40,532 lb.

When steel angles are used, the head of a boiler, thirty-six (36) inches in diameter, designed for one hundred (100) pounds working pressure, shall be stayed

by two (2) six by three and one-half by one-half ($6 \times 3\ 1/2 \times 1/2$) inch steel angles, as shown in the figure or by other sized commercial steel angles the resistance of which shall be equal to or greater than the maximum bending moment.

Distance from tubes to shell = 15 1/2 in.

Area to be stayed = 220 sq. in.

Load at 100 lb. pressure = 22,000 lb.

$$\frac{WL}{8} = \frac{22000 \times 27}{8} = 74{,}250 \text{ lb.} \qquad \text{Moment of inertia} =$$

$$I = 1/12 \times 6^3 \times 1/2 = 9.$$
$$y = 6 \div 2 = 3.$$

$$\frac{fI}{y} = M = \frac{16000 \times 9}{3} = 48{,}000 \text{ lb. for one angle.}$$

Resistance of one angle = 48,000 lb.

Resistance of two angles = 96,000 lb.

PITCH OF STAY BOLTS ON FURNACE SHEETS

The longitudinal pitch between stay bolts on the furnace sheet of an internally fired boiler, in which the external diameter of the furnace is thirty-eight (38) inches or less, except a corrugated furnace or a furnace strengthened by an Adamson ring or equivalent, shall not exceed that given by the following formula:

$$L = \left(\frac{ct^2}{Pd}\right)^2, \quad t = \sqrt{\frac{Pd\sqrt{L}}{C}},$$

$$P = \frac{ct^2}{d\sqrt{L}}, \quad d = \frac{ct^2}{P\sqrt{L}}$$

L = longitudinal pitch of stay bolts, in inches, or one-half the height of furnace when only one circumferential row of stay bolts is required.

C = a constant = 110.

t = thickness of furnace sheet, in *thirty-seconds* of an inch.

P = working pressure per square inch in pounds.

d = external diameter of furnace in inches.

TABLE I.—AREAS AND CIRCUMFERENCES OF CIRCLES FROM 1 TO 100

Dia.	Area	Circum.	Dia.	Area	Circum.	Dia.	Area	Circum.
$\frac{1}{32}$	0.00077	0.098175	2	3.1416	6.28319	5	19.635	15.7080
$\frac{3}{64}$	0.00173	0.147262	$\frac{1}{16}$	3.3410	6.47953	$\frac{1}{16}$	20.129	15.9043
$\frac{1}{16}$	0.00307	0.196350	$\frac{1}{8}$	3.5466	6.67588	$\frac{1}{8}$	20.629	16.1007
$\frac{3}{32}$	0.00690	0.294524	$\frac{3}{16}$	3.7583	6.87223	$\frac{3}{16}$	21.135	16.2970
$\frac{1}{8}$	0.01227	0.392699	$\frac{1}{4}$	3.9761	7.06858	$\frac{1}{4}$	21.648	16.4934
$\frac{5}{32}$	0.01917	0.490874	$\frac{5}{16}$	4.2000	7.26493	$\frac{5}{16}$	22.166	16.6897
$\frac{3}{16}$	0.02761	0.589049	$\frac{3}{8}$	4.4301	7.46128	$\frac{3}{8}$	22.691	16.8861
$\frac{7}{32}$	0.03758	0.687223	$\frac{7}{16}$	4.6664	7.65763	$\frac{7}{16}$	23.221	17.0824
$\frac{1}{4}$	0.04909	0.785398	$\frac{1}{2}$	4.9087	7.85398	$\frac{1}{2}$	23.758	17.2788
$\frac{9}{32}$	0.06213	0.883573	$\frac{9}{16}$	5.1572	8.05033	$\frac{9}{16}$	24.301	17.4751
$\frac{5}{16}$	0.07670	0.981748	$\frac{5}{8}$	5.4119	8.24668	$\frac{5}{8}$	24.850	17.6715
$\frac{11}{32}$	0.09281	1.07992	$\frac{11}{16}$	5.6727	8.44303	$\frac{11}{16}$	25.406	17.8678
$\frac{3}{8}$	0.11045	1.17810	$\frac{3}{4}$	5.9396	8.63938	$\frac{3}{4}$	25.967	18.0642
$\frac{13}{32}$	0.12962	1.27627	$\frac{13}{16}$	6.2126	8.83573	$\frac{13}{16}$	26.535	18.2605
$\frac{7}{16}$	0.15033	1.37445	$\frac{7}{8}$	6.4918	9.03208	$\frac{7}{8}$	27.109	18.4569
$\frac{15}{32}$	0.17257	1.47262	$\frac{15}{16}$	6.7771	9.22843	$\frac{15}{16}$	27.688	18.6532
$\frac{1}{2}$	0.19635	1.57080	3	7.0686	9.42478	6	28.274	18.8496
$\frac{17}{32}$	0.22166	1.66897	$\frac{1}{16}$	7.3662	9.62113	$\frac{1}{8}$	29.465	19.2423
$\frac{9}{16}$	0.24850	1.76715	$\frac{1}{8}$	7.6699	9.81748	$\frac{1}{4}$	30.680	19.6350
$\frac{19}{32}$	0.27688	1.86532	$\frac{3}{16}$	7.9798	10.0138	$\frac{3}{8}$	31.919	20.0277
$\frac{5}{8}$	0.30680	1.96350	$\frac{1}{4}$	8.2958	10.2102	$\frac{1}{2}$	33.183	20.4204
$\frac{21}{32}$	0.33824	2.06167	$\frac{5}{16}$	8.6179	10.4065	$\frac{5}{8}$	34.472	20.8131
$\frac{11}{16}$	0.37122	2.15984	$\frac{3}{8}$	8.9462	10.6029	$\frac{3}{4}$	35.785	21.2058
$\frac{23}{32}$	0.40574	2.25802	$\frac{7}{16}$	9.2806	10.7992	$\frac{7}{8}$	37.122	21.5984
$\frac{3}{4}$	0.44179	2.35619	$\frac{1}{2}$	9.6211	10.9956	7	38.485	21.9911
$\frac{25}{32}$	0.47937	2.45437	$\frac{9}{16}$	9.9678	11.1919	$\frac{1}{8}$	39.871	22.3838
$\frac{13}{16}$	0.51849	2.55254	$\frac{5}{8}$	10.321	11.3883	$\frac{1}{4}$	41.282	22.7765
$\frac{27}{32}$	0.55914	2.65072	$\frac{11}{16}$	10.680	11.5846	$\frac{3}{8}$	42.718	23.1692
$\frac{7}{8}$	0.60132	2.74889	$\frac{3}{4}$	11.045	11.7810	$\frac{1}{2}$	44.179	23.5619
$\frac{29}{32}$	0.64504	2.84707	$\frac{13}{16}$	11.416	11.9773	$\frac{5}{8}$	45.664	23.9546
$\frac{15}{16}$	0.69029	2.94524	$\frac{7}{8}$	11.793	12.1737	$\frac{3}{4}$	47.173	24.3473
$\frac{31}{32}$	0.73708	3.04342	$\frac{15}{16}$	12.177	12.3700	$\frac{7}{8}$	48.707	24.7400
1	0.78540	3.14159	4	12.566	12.5664	8	50.265	25.1327
$\frac{1}{16}$	0.88664	3.33794	$\frac{1}{16}$	12.962	12.7627	$\frac{1}{8}$	51.849	25.5224
$\frac{1}{8}$	0.99402	3.53429	$\frac{1}{8}$	13.364	12.9591	$\frac{1}{4}$	53.456	25.9181
$\frac{3}{16}$	1.1075	3.73064	$\frac{3}{16}$	13.772	13.1554	$\frac{3}{8}$	55.088	26.3108
$\frac{1}{4}$	1.2272	3.92699	$\frac{1}{4}$	14.186	13.3518	$\frac{1}{2}$	56.745	26.7035
$\frac{5}{16}$	1.3530	4.12334	$\frac{5}{16}$	14.607	13.5481	$\frac{5}{8}$	58.426	27.0962
$\frac{3}{8}$	1.4849	4.31969	$\frac{3}{8}$	15.033	13.7445	$\frac{3}{4}$	60.132	27.4889
$\frac{7}{16}$	1.6230	4.51604	$\frac{7}{16}$	15.466	13.9408	$\frac{7}{8}$	61.862	27.8816
$\frac{1}{2}$	1.7671	4.71239	$\frac{1}{2}$	15.904	14.1372	9	63.617	28.2743
$\frac{9}{16}$	1.9175	4.90874	$\frac{9}{16}$	16.349	14.3335	$\frac{1}{8}$	65.397	28.6670
$\frac{5}{8}$	2.0739	5.10509	$\frac{5}{8}$	16.800	14.5299	$\frac{1}{4}$	67.201	29.0597
$\frac{11}{16}$	2.2365	5.30144	$\frac{11}{16}$	17.257	14.7262	$\frac{3}{8}$	69.029	29.4524
$\frac{3}{4}$	2.4053	5.49779	$\frac{3}{4}$	17.721	14.9226	$\frac{1}{2}$	70.882	29.8451
$\frac{13}{16}$	2.5802	5.69414	$\frac{13}{16}$	18.190	15.1189	$\frac{5}{8}$	72.760	30.2378
$\frac{7}{8}$	2.7612	5.89049	$\frac{7}{8}$	18.665	15.3153	$\frac{3}{4}$	74.662	30.6305
$\frac{15}{16}$	2.9483	6.08684	$\frac{15}{16}$	19.147	15.5116	$\frac{7}{8}$	76.589	31.0232

TABLE I.—AREAS AND CIRCUMFERENCES OF CIRCLES FROM 1 TO 100 (*Continued*)

Dia.	Area	Circum.	Dia.	Area	Circum.	Dia.	Area	Circum.
10	78.540	31.4159	16	201.06	50.2655	22	380.13	69.1150
1/8	80.516	31.8086	1/8	204.22	50.6582	1/8	384.46	69.5077
1/4	82.516	32.2013	1/4	207.39	51.0509	1/4	388.82	69.9004
3/8	84.541	32.5940	3/8	210.60	51.4436	3/8	393.20	70.2931
1/2	86.590	32.9867	1/2	213.82	51.8363	1/2	397.61	70.6858
5/8	88.664	33.3794	5/8	217.08	52.2290	5/8	402.04	71.0785
3/4	90.763	33.7721	3/4	220.35	52.6217	3/4	406.49	71.4712
7/8	92.886	34.1648	7/8	223.65	53.0144	7/8	410.97	71.8639
11	95.033	34.5575	17	226.98	53.4071	23	415.48	72.2566
1/8	97.205	34.9502	1/8	230.33	53.7998	1/8	420.00	72.6493
1/4	99.402	35.3429	1/4	233.71	54.1925	1/4	424.56	73.0420
3/8	101.62	35.7356	3/8	237.10	54.5852	3/8	429.13	73.4347
1/2	103.87	36.1283	1/2	240.53	54.9779	1/2	433.74	73.8274
5/8	106.14	36.5210	5/8	243.98	55.3706	5/8	438.36	74.2201
3/4	108.43	36.9137	3/4	247.45	55.7633	3/4	443.01	74.6128
7/8	110.75	37.3064	7/8	250.95	56.1560	7/8	447.69	75.0055
12	113.10	37.6991	18	254.47	56.5487	24	452.39	75.3982
1/8	115.47	38.0918	1/8	258.02	56.9414	1/8	457.11	75.7909
1/4	117.86	38.4845	1/4	261.59	57.3341	1/4	461.86	76.1836
3/8	120.28	38.8772	3/8	265.18	57.7268	3/8	466.64	76.5783
1/2	122.72	39.2699	1/2	268.80	58.1195	1/2	471.44	76.9690
5/8	125.19	39.6626	5/8	272.45	58.5122	5/8	476.26	77.3617
3/4	127.68	40.0553	3/4	276.12	58.9049	3/4	481.11	77.7544
7/8	130.19	40.4480	7/8	279.81	59.2976	7/8	485.98	78.1471
13	132.73	40.8407	19	283.53	59.6903	25	490.87	78.5398
1/8	135.30	41.2334	1/8	287.27	60.0830	1/8	495.79	78.9325
1/4	137.89	41.6261	1/4	291.04	60.4757	1/4	500.74	79.3252
3/8	140.50	42.0188	3/8	294.83	60.8684	3/8	505.71	79.7179
1/2	143.14	42.4115	1/2	298.65	61.2611	1/2	510.71	80.1105
5/8	145.80	42.8042	5/8	302.49	61.6538	5/8	515.72	80.5033
3/4	148.49	43.1969	3/4	306.35	62.0465	3/4	520.77	80.8960
7/8	151.20	43.5896	7/8	310.24	62.4392	7/8	525.84	81.2887
14	153.94	43.9823	20	314.16	62.8319	26	530.93	81.6814
1/8	156.70	44.3750	1/8	318.10	63.2246	1/8	536.05	82.0741
1/4	159.48	44.7677	1/4	322.06	63.6173	1/4	541.19	82.4668
3/8	162.30	45.1604	3/8	326.05	64.0100	3/8	546.35	82.8595
1/2	165.13	45.5531	1/2	330.06	64.4026	1/2	551.55	83.2522
5/8	167.99	45.9458	5/8	334.10	64.7953	5/8	556.76	83.6449
3/4	170.87	46.3385	3/4	338.16	65.1880	3/4	562.00	84.0376
7/8	173.78	46.7312	7/8	342.25	65.5807	7/8	567.27	84.4303
15	176.71	47.1239	21	346.36	65.9734	27	572.56	84.8230
1/8	179.67	47.5166	1/8	350.50	66.3661	1/8	577.87	85.2157
1/4	182.65	47.9093	1/4	354.66	66.7588	1/4	583.21	85.6084
3/8	185.66	48.3020	3/8	358.84	67.1515	3/8	588.57	86.0011
1/2	188.69	48.6947	1/2	363.05	67.5442	1/2	593.96	86.3938
5/8	191.75	49.0874	5/8	367.28	67.9369	5/8	599.37	86.7865
3/4	194.83	49.4801	3/4	371.54	68.3296	3/4	604.81	87.1792
7/8	197.93	49.8728	7/8	375.83	68.7223	7/8	610.27	87.5719

TABLE I.—AREAS AND CIRCUMFERENCES OF CIRCLES FROM 1 TO 100 (*Continued*)

Dia.	Area	Circum.	Dia.	Area	Circum.	Dia.	Area	Circum.
28	615.75	87.9646	34	907.92	106.814	40	1256.6	125.664
1/8	621.26	88.3573	1/8	914.61	107.207	1/8	1264.5	126.056
1/4	626.80	88.7500	1/4	921.32	107.600	1/4	1272.4	126.449
3/8	632.36	89.1427	3/8	928.06	107.992	3/8	1280.3	126.842
1/2	637.94	89.5354	1/2	934.82	108.385	1/2	1288.2	127.235
5/8	643.55	89.9281	5/8	941.61	108.788	5/8	1296.2	127.627
3/4	649.18	90.3208	3/4	948.42	109.170	3/4	1304.2	128.020
7/8	656.84	90.7135	7/8	955.25	109.563	7/8	1312.2	128.413
29	660.52	91.1062	35	962.11	109.956	41	1320.3	128.805
1/8	666.23	91.4989	1/8	969.00	110.348	1/8	1328.3	129.198
1/4	671.96	91.8916	1/4	975.91	110.741	1/4	1336.4	129.591
3/8	677.71	92.2843	3/8	982.84	111.134	3/8	1344.5	129.993
1/2	683.49	92.6770	1/2	989.80	111.527	1/2	1352.7	130.376
5/8	689.30	93.0697	5/8	996.78	111.919	5/8	1360.8	130.769
3/4	695.13	93.4624	3/4	1003.8	112.312	3/4	1369.0	131.161
7/8	700.98	93.8551	7/8	1010.8	112.705	7/8	1377.2	131.554
30	706.86	94.2478	36	1017.9	113.097	42	1385.4	131.947
1/8	712.76	94.6405	1/8	1025.0	113.490	1/8	1393.7	132.340
1/4	718.69	95.0332	1/4	1032.1	113.883	1/4	1402.0	132.732
3/8	724.64	95.4259	3/8	1039.2	114.275	3/8	1410.3	133.125
1/2	730.62	95.8186	1/2	1046.3	114.668	1/2	1418.6	133.518
5/8	736.62	96.2113	5/8	1053.5	115.061	5/8	1427.0	133.910
3/4	742.64	96.6040	3/4	1060.7	115.454	3/4	1435.4	134.303
7/8	748.69	96.9967	7/8	1068.0	115.846	7/8	1443.8	134.696
31	754.77	97.3894	37	1075.2	116.239	43	1452.2	135.088
1/8	760.87	97.7821	1/8	1082.5	116.632	1/8	1460.7	135.481
1/4	766.99	98.1748	1/4	1089.8	117.024	1/4	1469.1	135.874
3/8	773.14	98.5675	3/8	1097.1	117.417	3/8	1477.6	136.267
1/2	779.31	98.9602	1/2	1104.5	117.810	1/2	1486.2	136.659
5/8	785.51	99.3529	5/8	1111.8	118.202	5/8	1494.7	137.052
3/4	791.73	99.7456	3/4	1119.2	118.596	3/4	1503.3	137.445
7/8	797.98	100.138	7/8	1126.7	118.988	7/8	1511.9	137.837
32	804.25	100.531	38	1134.1	119.381	44	1520.5	138.230
1/8	810.54	100.924	1/8	1141.6	119.773	1/8	1529.2	138.623
1/4	816.86	101.316	1/4	1149.1	120.166	1/4	1537.9	139.015
3/8	823.21	101.709	3/8	1156.6	120.559	3/8	1546.6	139.408
1/2	829.58	102.102	1/2	1164.2	120.951	1/2	1555.3	139.801
5/8	835.97	102.494	5/8	1171.7	121.344	5/8	1564.0	140.194
3/4	842.39	102.887	3/4	1179.3	121.737	3/4	1572.8	140.586
7/8	848.83	103.280	7/8	1186.9	122.129	7/8	1581.6	140.979
33	855.30	103.673	39	1194.6	122.522	45	1590.4	141.372
1/8	861.79	104.065	1/8	1202.3	122.915	1/8	1599.3	141.764
1/4	868.31	104.458	1/4	1210.0	123.308	1/4	1608.2	142.157
3/8	874.85	104.851	3/8	1217.7	123.700	3/8	1617.0	142.550
1/2	881.41	105.243	1/2	1225.4	124.093	1/2	1626.0	142.942
5/8	888.00	105.636	5/8	1233.2	124.486	5/8	1634.9	143.335
3/4	894.62	106.029	3/4	1241.0	124.878	3/4	1643.9	143.728
7/8	901.26	106.421	7/8	1248.8	125.271	7/8	1652.9	144.121

TABLE I.—AREAS AND CIRCUMFERENCES OF CIRCLES FROM 1 TO 100 (*Continued*)

Dia.	Area	Circum.	Dia.	Area	Circum.	Dia.	Area	Circum.
46	1661.9	144.513	52	2123.7	163.363	58	2642.1	182.212
$\frac{1}{8}$	1670.9	144.906	$\frac{1}{8}$	2133.9	163.756	$\frac{1}{8}$	2653.5	182.605
$\frac{1}{4}$	1680.0	145.299	$\frac{1}{4}$	2144.2	164.143	$\frac{1}{4}$	2664.9	182.998
$\frac{3}{8}$	1689.1	145.691	$\frac{3}{8}$	2154.5	164.541	$\frac{3}{8}$	2676.4	183.390
$\frac{1}{2}$	1698.2	146.084	$\frac{1}{2}$	2164.8	164.934	$\frac{1}{2}$	2687.8	183.783
$\frac{5}{8}$	1707.4	146.477	$\frac{5}{8}$	2175.1	165.326	$\frac{5}{8}$	2699.3	184.176
$\frac{3}{4}$	1716.5	146.869	$\frac{3}{4}$	2185.4	165.719	$\frac{3}{4}$	2710.9	184.569
$\frac{7}{8}$	1725.7	147.262	$\frac{7}{8}$	2195.8	166.112	$\frac{7}{8}$	2722.4	184.961
47	1734.9	147.655	53	2206.2	166.504	59	2734.0	185.354
$\frac{1}{8}$	1744.2	148.048	$\frac{1}{8}$	2216.6	166.897	$\frac{1}{8}$	2745.6	185.747
$\frac{1}{4}$	1753.5	148.440	$\frac{1}{4}$	2227.0	167.290	$\frac{1}{4}$	2757.2	186.139
$\frac{3}{8}$	1762.7	148.833	$\frac{3}{8}$	2237.5	167.683	$\frac{3}{8}$	2768.8	186.532
$\frac{1}{2}$	1772.1	149.226	$\frac{1}{2}$	2248.0	168.075	$\frac{1}{2}$	2780.5	186.925
$\frac{5}{8}$	1781.4	149.618	$\frac{5}{8}$	2258.5	168.468	$\frac{5}{8}$	2792.2	187.317
$\frac{3}{4}$	1790.8	150.011	$\frac{3}{4}$	2269.1	168.861	$\frac{3}{4}$	2803.9	187.710
$\frac{7}{8}$	1800.1	150.404	$\frac{7}{8}$	2279.6	169.253	$\frac{7}{8}$	2815.7	188.103
48	1809.6	150.796	54	2290.2	169.646	60	2827.4	188.496
$\frac{1}{8}$	1819.0	151.189	$\frac{1}{8}$	2300.8	170.039	$\frac{1}{8}$	2839.2	188.888
$\frac{1}{4}$	1828.5	151.582	$\frac{1}{4}$	2311.5	170.431	$\frac{1}{4}$	2851.0	189.281
$\frac{3}{8}$	1837 9	151.975	$\frac{3}{8}$	2322.1	170.824	$\frac{3}{8}$	2862.9	189.674
$\frac{1}{2}$	1847.5	152.367	$\frac{1}{2}$	2332.8	171.217	$\frac{1}{2}$	2874.8	190.066
$\frac{5}{8}$	1857.0	152.760	$\frac{5}{8}$	2343.5	171.609	$\frac{5}{8}$	2886.6	190.459
$\frac{3}{4}$	1866.5	153.153	$\frac{3}{4}$	2354.3	172.002	$\frac{3}{4}$	2898.6	190.852
$\frac{7}{8}$	1876.1	153.544	$\frac{7}{8}$	2365.0	172.395	$\frac{7}{8}$	2910.5	191.244
49	1885.7	153.938	55	2375.8	172.788	61	2922.5	191.637
$\frac{1}{8}$	1895.4	154.331	$\frac{1}{8}$	2386.6	173.180	$\frac{1}{8}$	2934.5	192.030
$\frac{1}{4}$	1905.0	154.723	$\frac{1}{4}$	2397.5	173.573	$\frac{1}{4}$	2946.5	192.423
$\frac{3}{8}$	1914.7	155.116	$\frac{3}{8}$	2408.3	173.966	$\frac{3}{8}$	2958.5	192.815
$\frac{1}{2}$	1924.2	155.509	$\frac{1}{2}$	2419.2	174.358	$\frac{1}{2}$	2970.6	193.208
$\frac{5}{8}$	1934.2	155.904	$\frac{5}{8}$	2430.1	174.751	$\frac{5}{8}$	2982.7	193.601
$\frac{3}{4}$	1943.9	156.294	$\frac{3}{4}$	2441.1	175.144	$\frac{3}{4}$	2994.8	193.993
$\frac{7}{8}$	1953.7	156.687	$\frac{7}{8}$	2452.0	175.536	$\frac{7}{8}$	3006.9	194.386
50	1963.5	157.080	56	2463.0	175.929	62	3019.1	194.779
$\frac{1}{8}$	1973.3	157.472	$\frac{1}{8}$	2474.0	176.322	$\frac{1}{8}$	3031.3	195.171
$\frac{1}{4}$	1983.2	157.865	$\frac{1}{4}$	2485.0	176.715	$\frac{1}{4}$	3043.5	195.564
$\frac{3}{8}$	1993.1	158.258	$\frac{3}{8}$	2496.1	177.107	$\frac{3}{8}$	3055.7	195.957
$\frac{1}{2}$	2003.0	158.650	$\frac{1}{2}$	2507.2	177.500	$\frac{1}{2}$	3068.0	196.350
$\frac{5}{8}$	2012.9	159.043	$\frac{5}{8}$	2518.3	177.893	$\frac{5}{8}$	3080.3	196.742
$\frac{3}{4}$	2022.8	159.436	$\frac{3}{4}$	2529.4	178.285	$\frac{3}{4}$	3092.6	197.135
$\frac{7}{8}$	2032.8	159.829	$\frac{7}{8}$	2540.6	178.678	$\frac{7}{8}$	3104.9	197.528
51	2042.8	160.221	57	2551.8	179.071	63	3117.2	197.920
$\frac{1}{8}$	2052.8	160.614	$\frac{1}{8}$	2563.0	179.463	$\frac{1}{8}$	3129.6	198.313
$\frac{1}{4}$	2062.9	161.007	$\frac{1}{4}$	2574.2	179.856	$\frac{1}{4}$	3142.0	198.706
$\frac{3}{8}$	2073.0	161.399	$\frac{3}{8}$	2585.4	180.249	$\frac{3}{8}$	3154.5	199.098
$\frac{1}{2}$	2083.1	161.792	$\frac{1}{2}$	2596.7	180.642	$\frac{1}{2}$	3166.9	199.491
$\frac{5}{8}$	2093.2	162.185	$\frac{5}{8}$	2608.0	181.034	$\frac{5}{8}$	3179.4	199.884
$\frac{3}{4}$	2103.3	162.577	$\frac{3}{4}$	2619.4	181.427	$\frac{3}{4}$	3191.9	200.277
$\frac{7}{8}$	2113.5	162.970	$\frac{7}{8}$	2630.7	181.820	$\frac{7}{8}$	3204.4	200.669

TABLE I.—AREAS AND CIRCUMFERENCES OF CIRCLES FROM 1 TO 100 (*Continued*)

Dia.	Area	Circum.	Dia.	Area	Circum.	Dia.	Area	Circum.
64	3217.0	201.062	70	3848.5	219.911	76	4536.5	238.761
1/8	3229.6	201.455	1/8	3862.2	220.304	1/8	4551.4	239.154
1/4	3242.2	201.847	1/4	3876.0	220.697	1/4	4566.4	239.546
3/8	3254.8	202.240	3/8	3889.8	221.090	3/8	4581.3	239.939
1/2	3267.5	202.633	1/2	3903.6	221.482	1/2	4596.3	240 332
5/8	3280.1	203.025	5/8	3917.5	221.875	5/8	4611.4	240.725
3/4	3292.8	203.418	3/4	3931.4	222.268	3/4	4626.4	241.117
7/8	3305.6	203.811	7/8	3945.3	222.660	7/8	4641.5	241.510
65	3318.3	204.204	71	3959.2	223.053	77	4656.6	241.903
1/8	3331.1	204.596	1/8	3973.1	223.446	1/8	4671.8	242.295
1/4	3343.9	204.989	1/4	3987.1	223.838	1/4	4686.9	242.688
3/8	3356.7	205.382	3/8	4001.1	224.231	3/8	4702.1	243.081
1/2	3369.6	205.774	1/2	4015.2	224.624	1/2	4717.3	243.473
5/8	3382.4	206.167	5/8	4029.2	225.017	5/8	4732.5	243.866
3/4	3395.3	206.560	3/4	4043.3	225.409	3/4	4747.8	244.259
7/8	3408.2	206.952	7/8	4057.4	225.802	7/8	4763.1	244.652
66	3421.2	207.345	72	4071.5	226.195	78	4778.4	245.044
1/8	3434.3	207.738	1/8	4085.7	226.587	1/8	4793.7	245.437
1/4	3447.2	208.131	1/4	4099.8	226.930	1/4	4809.0	245.830
3/8	3460.2	208.523	3/8	4114.0	227.373	3/8	4824.4	246.222
1/2	3473.2	208.916	1/2	4128.2	227.765	1/2	4839.8	246.615
5/8	3486.3	209.309	5/8	4142.5	228.158	5/8	4855.2	247.008
3/4	3499.4	209.701	3/4	4156.8	228.551	3/4	4870.7	247.400
7/8	3512.5	210.094	7/8	4171.1	228.944	7/8	4886.2	247.793
67	3525.7	210.487	73	4185.4	229.336	79	4901.7	248.186
1/8	3538.8	210.879	1/8	4199.7	229.729	1/8	4917.2	248.579
1/4	3552.0	211.272	1/4	4214.1	230.122	1/4	4932.7	248.971
3/8	3565.2	211.665	3/8	4228.5	230.514	3/8	4948.3	249.364
1/2	3578.5	212.058	1/2	4242.9	230.907	1/2	4963.9	249.757
5/8	3591.7	212.450	5/8	4257.4	231.300	5/8	4979.5	250.149
3/4	3605.0	212.843	3/4	4271.8	231.692	3/4	4995.2	250.542
7/8	3618.3	213.236	7/8	4286.3	232.085	7/8	5010.9	250.935
68	3631.7	213.628	74	4300.8	232.478	80	5026.5	251.327
1/8	3645.0	214.021	1/8	4315.4	232.871	1/8	5042.3	251.720
1/4	3658.4	214.414	1/4	4329.9	233.263	1/4	5058.0	252.113
3/8	3671.8	214.806	3/8	4344.5	233.656	3/8	5073.8	252.506
1/2	3685.3	215.199	1/2	4359.2	234.049	1/2	5089.6	252.898
5/8	3698.7	215.592	5/8	4373.8	234.441	5/8	5105.4	253.291
3/4	3712.2	215.984	3/4	4388.5	234.334	3/4	5121.2	253.684
7/8	3725.7	216.337	7/8	4403.1	235.227	7/8	5137.1	254.076
69	3739.3	216.770	75	4417.9	235.619	81	5153.0	254.469
1/8	3752.8	217.163	1/8	4432.6	236.012	1/8	5168.9	254.862
1/4	3766.4	217.555	1/4	4447.4	236.405	1/4	5184.9	255.254
3/8	3780.0	217.948	3/8	4462.2	236.798	3/8	5200.8	255.647
1/2	3793.7	218.341	1/2	4477.0	237.190	1/2	5216.8	256.040
5/8	3807.3	218.733	5/8	4491.8	237.583	5/8	5232.8	256.433
3/4	3821.0	219.126	3/4	4506.7	237.976	3/4	5248.9	256.825
7/8	3834.7	219.519	7/8	4521.5	238.368	7/8	5264.9	257.213

TABLE I.—AREAS AND CIRCUMFERENCES OF CIRCLES FROM 1 TO 100 (*Continued*)

Dia.	Area	Circum.	Dia.	Area	Circum.	Dia.	Area	Circum.
82	5281.0	257.611	88	6082.1	276.460	94	6939.8	295.310
1/8	5297.1	258.003	1/8	6099.4	276.853	1/8	6958.2	295.702
1/4	5313.3	258.396	1/4	6256.7	277.846	1/4	6976.7	296.095
3/8	5329.4	258.789	3/8	6134.1	277.638	3/8	6995.3	296.488
1/2	5345.6	259.181	1/2	6151.4	278.031	1/2	7013.8	296.881
5/8	5361.8	259.574	5/8	6168.8	278.424	5/8	7032.4	297.273
3/4	5378.1	259.967	3/4	6186.2	278.816	3/4	7051.0	297.666
7/8	5394.3	260.359	7/8	6203.7	279.209	7/8	7069.6	298.059
83	5410.6	260.752	89	6221.1	279.602	95	7088.2	298.451
1/8	5426.9	261.145	1/8	6238.6	279.994	1/8	7106.9	298.844
1/4	5443.3	261.538	1/4	6256.1	280.387	1/4	7125.6	299.237
3/8	5459.6	261.930	3/8	6273.7	280.780	3/8	7144.3	299.629
1/2	5476.0	262.323	1/2	6291.2	281.173	1/2	7163.0	300.022
5/8	5492.4	262.716	5/8	6308.8	281.565	5/8	7181.8	300.415
3/4	5508.8	263.103	3/4	6326.4	281.958	3/4	7200.6	300.807
7/8	5525.3	263.501	7/8	6344.1	282.351	7/8	7219.4	301.200
84	5541.8	263.894	90	6361.7	282.743	96	7238.2	301.593
1/8	5558.3	264.286	1/8	6379.4	283.136	1/8	7257.1	301.986
1/4	5574.8	264.679	1/4	6397.1	283.529	1/4	7276.0	302.378
3/8	5591.4	265.072	3/8	6414.9	283.921	3/8	7294.9	302.771
1/2	5607.9	265.465	1/2	6432.6	284.314	1/2	7313.8	303.164
5/8	5624.5	265.857	5/8	6450.4	284.707	5/8	7332.8	303.556
3/4	5641.2	266.250	3/4	6468.2	285.100	3/4	7351.8	303.949
7/8	5657.8	266.643	7/8	6486.0	285.492	7/8	7370.8	304.342
85	5674.5	267.035	91	6503.9	285.885	97	7389.8	304.734
1/8	5691.2	267.428	1/8	6521.8	286.278	1/8	7408.9	305.127
1/4	5707.9	267.821	1/4	6539.7	286.670	1/4	7428.0	305.520
3/8	5724.7	268.213	3/8	6557.6	287.063	3/8	7447.1	305.913
1/2	5741.5	268.606	1/2	6575.5	287.456	1/2	7466.2	306.305
5/8	5758.3	268.999	5/8	6593.5	287.848	5/8	7485.3	306.698
3/4	5775.1	269.392	3/4	6611.5	288.241	3/4	7504.5	307.091
7/8	5791.9	269.784	7/8	6629.6	288.634	7/8	7523.7	307.483
86	5808.8	270.177	92	6647.6	289.027	98	7543.0	307.876
1/8	5825.7	270.570	1/8	6665.7	289.419	1/8	7562.2	308.269
1/4	5842.6	270.962	1/4	6683.8	289.812	1/4	7581.5	308.661
3/8	5859.6	271.355	3/8	6701.9	290.205	3/8	7600.8	309.064
1/2	5876.5	271.748	1/2	6720.1	290.597	1/2	7620.1	309.447
5/8	5893.5	272.140	5/8	6738.2	290.990	5/8	7639.5	309.840
3/4	5910.6	272.533	3/4	6756.4	291.383	3/4	7658.9	310.232
7/8	5927.6	272.926	7/8	6774.7	291.775	7/8	7678.3	310.625
87	5944.7	273.319	93	6792.9	292.168	99	7697.7	311.018
1/8	5961.8	273.711	1/8	6811.2	292.561	1/8	7717.1	311.410
1/4	5978.9	274.104	1/4	6829.5	292.954	1/4	7736.6	311.803
3/8	5996.0	274.497	3/8	6847.8	293.346	3/8	7756.1	312.196
1/2	6013.2	274.889	1/2	6866.1	293.739	1/2	7775.6	312.588
5/8	6030.4	275.282	5/8	6884.5	294.132	5/8	7795.2	312.981
3/4	6047.6	275.675	3/4	6902.9	294.524	3/4	7814.8	313.374
7/8	6064.9	276.067	7/8	6921.3	294.917	7/8	7834.4	313.767

TABLE II.—DECIMAL EQUIVALENTS OF FRACTIONS OF AN INCH. (ADVANCING BY 8THS, 16THS, 32NDS AND 64THS.)

8ths	32nds	64ths	64ths
$\frac{1}{8}$ = .125	$\frac{1}{32}$ = .03125	$\frac{1}{64}$ = .015625	$\frac{33}{64}$ = .515625
$\frac{1}{4}$ = .250	$\frac{3}{32}$ = .09375	$\frac{3}{64}$ = .046875	$\frac{35}{64}$ = .546875
$\frac{3}{8}$ = .375	$\frac{5}{32}$ = .15625	$\frac{5}{64}$ = .078125	$\frac{37}{64}$ = .578125
$\frac{1}{2}$ = .500	$\frac{7}{32}$ = .21875	$\frac{7}{64}$ = .109375	$\frac{39}{64}$ = .609375
$\frac{5}{8}$ = .625	$\frac{9}{32}$ = .28125	$\frac{9}{64}$ = .140625	$\frac{41}{64}$ = .640625
$\frac{3}{4}$ = .750	$\frac{11}{32}$ = .34375	$\frac{11}{64}$ = .171875	$\frac{43}{64}$ = .671875
$\frac{7}{8}$ = .875	$\frac{13}{32}$ = .40625	$\frac{13}{64}$ = .203125	$\frac{45}{64}$ = .703125
	$\frac{15}{32}$ = .46875	$\frac{15}{64}$ = .234375	$\frac{47}{64}$ = .734375
16ths.			
$\frac{1}{16}$ = .0625	$\frac{17}{32}$ = .53125	$\frac{17}{64}$ = .265625	$\frac{49}{64}$ = .765625
$\frac{3}{16}$ = .1875	$\frac{19}{32}$ = .59375	$\frac{19}{64}$ = .296875	$\frac{51}{64}$ = .796875
$\frac{5}{16}$ = .3125	$\frac{21}{32}$ = .65625	$\frac{21}{64}$ = .328125	$\frac{53}{64}$ = .828125
$\frac{7}{16}$ = .4375	$\frac{23}{32}$ = .71875	$\frac{23}{64}$ = .359375	$\frac{55}{64}$ = .859375
$\frac{9}{16}$ = .5625	$\frac{25}{32}$ = .78125	$\frac{25}{64}$ = .390625	$\frac{57}{64}$ = .890625
$\frac{11}{16}$ = .6875	$\frac{27}{32}$ = .84375	$\frac{27}{64}$ = .421875	$\frac{59}{64}$ = .921875
$\frac{13}{16}$ = .8125	$\frac{29}{32}$ = .90625	$\frac{29}{64}$ = .453125	$\frac{61}{64}$ = .953125
$\frac{15}{16}$ = .9375	$\frac{31}{32}$ = .96875	$\frac{31}{64}$ = .484375	$\frac{63}{64}$ = .984375

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
1	1	1	1.0000	1.0000	3.142	0.7854
2	4	8	1.4142	1.2599	6.283	3.1416
3	9	27	1.7321	1.4422	9.425	7.0686
4	16	64	2.0000	1.5874	12.566	12.5664
5	25	125	2.2361	1.7100	15.708	19.6350
6	36	216	2.4495	1.8171	18.850	28.2743
7	49	343	2.6458	1.9129	21.991	38.4845
8	64	512	2.8284	2.0000	25.133	50.2655
9	81	729	3.0000	2.0801	28.274	63.6173
10	100	1000	3.1623	2.1544	31.416	78.5398
11	121	1331	3.3166	2.2240	34.558	95.033
12	144	1728	3.4641	2.2894	37.699	113.097
13	169	2197	3.6056	2.3513	40.841	132.732
14	196	2744	3.7417	2.4101	43.982	153.938
15	225	3375	3.8730	2.4662	47.124	176.715
16	256	4096	4.0000	2.5198	50.265	201.062
17	289	4913	4.1231	2.5713	53.407	226.980
18	324	5832	4.2426	2.6207	56.549	254.469
19	361	6859	4.3589	2.6684	59.690	283.529
20	400	8000	4.4721	2.7144	62.832	314.159
21	441	9261	4.5826	2.7589	65.973	346.361
22	484	10648	4.6904	2.8020	69.115	380.133
23	529	12167	4.7958	2.8439	72.257	415.476
24	576	13824	4.8990	2.8845	75.398	452.389
25	625	15625	5.0000	2.9240	78.540	490.874
26	676	17576	5.0990	2.9625	81.681	530.929
27	729	19683	5.1962	3.0000	84.823	572.555
28	784	21952	5.2915	3.0366	87.965	615.752
29	841	24389	5.3852	3.0723	91.106	660.520
30	900	27000	5.4772	3.1072	94.248	706.858
31	961	29791	5.5678	3.1414	90.389	754.768
32	1024	32768	5.6569	3.1748	100.531	804.248
33	1089	35937	5.7446	3.2075	103.673	855.299
34	1156	39304	5.8310	3.2396	106.814	907.920
35	1225	42875	5.9161	3.2711	109.956	962.113
36	1296	46656	6.0000	3.3019	113.097	1017.88
37	1369	50653	6.0828	3.3322	116.239	1075.21
38	1444	54872	6.1644	3.3620	119.381	1134.11
39	1521	59319	6.2450	3.3912	122.522	1194.59
40	1600	64000	6.3246	3.4200	125.660	1256.64

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
41	1681	68921	6.4031	3.4482	128.81	1320.25
42	1764	74088	6.4807	3.4760	131.95	1385.44
43	1849	79507	6.5574	3.5034	135.09	1452.20
44	1936	85184	6.6332	3.5303	138.23	1520.53
45	2025	91125	6.7082	3.5569	141.37	1590.43
46	2116	97336	6.7823	3.5830	144.51	1661.90
47	2209	103823	6.8557	3.6088	147.65	1734.94
48	2304	110592	6.9282	3.6342	150.80	1809.56
49	2401	117649	7.0000	3.6593	153.94	1885.74
50	2500	125000	7.0711	3.6840	157.08	1963.50
51	2601	132651	7.1414	3.7084	160.22	2042.82
52	2704	140608	7.2111	3.7325	163.36	2123.72
53	2809	148877	7.2801	3.7563	166.50	2206.18
54	2916	157464	7.3485	3.7798	169.65	2290.22
55	3025	166375	7.4162	3.8030	172.79	2375.83
56	3136	175616	7.4833	3.8259	175.93	2463.01
57	3249	185193	7.5498	3.8485	179.07	2551.76
58	3364	195112	7.6158	3.8709	182.21	2642.08
59	3481	205379	7.6811	3.8930	185.35	2733.97
60	3600	216000	7.7460	3.9149	188.50	2827.43
61	3721	226981	7.8102	3.9365	191.64	2922.47
62	3844	238328	7.8740	3.9579	194.78	3019.07
63	3969	250047	7.9373	3.9791	197.92	3117.25
64	4096	262144	8.0000	4.0000	201.06	3216.99
65	4225	274625	8.0623	4.0207	204.20	3318.31
66	4356	287496	8.1240	4.0412	207.35	3421.19
67	4489	300763	8.1854	4.0615	210.49	3525.65
68	4624	314432	8.2462	4.0817	213.63	3631.68
69	4761	328509	8.3066	4.1016	216.77	3739.28
70	4900	343000	8.3666	4.1213	219.91	3848.45
71	5041	357911	8.4261	4.1408	223.05	3959.19
72	5184	373248	8.4853	4.1602	226.19	4071.50
73	5329	389017	8.5440	4.1793	229.34	4185.39
74	5476	405224	8.6023	4.1983	232.48	4300.84
75	5625	421875	8.6603	4.2172	235.62	4417.86
76	5776	438976	8.7178	4.2358	238.76	4536.46
77	5929	456533	8.7750	4.2543	241.90	4656.63
78	6084	474552	8.8318	4.2727	245.04	4778.36
79	6241	493039	8.8882	4.2908	248.19	4901.67
80	6400	512000	8.9443	4.3089	251.33	5026.55

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
81	6561	531441	9.0000	4.3267	254.47	5153.00
82	6724	551368	9.0554	4.3445	257.61	5281.02
83	6889	571787	9.1104	4.3621	260.75	5410.61
84	7056	592704	9.1652	4.3795	263.89	5541.77
85	7225	614125	9.2195	4.3968	267.04	5674.50
86	7396	636056	9.2736	4.4140	270.18	5808.80
87	7569	658503	9.3274	4.4310	273.32	5944.68
88	7744	681472	9.3808	4.4480	276.46	6082.12
89	7921	704969	9.4340	4.4647	279.60	6221.14
90	8100	729000	9.4868	4.4814	282.74	6361.73
91	8281	753571	9.5394	4.4979	285.88	6503.88
92	8464	778688	9.5917	4.5144	289.03	6647.61
93	8649	804357	9.6437	4.5307	292.17	6792.91
94	8836	830584	9.6954	4.5468	295.31	6939.78
95	9025	857375	9.7468	4.5629	298.45	7088.22
96	9216	884736	9.7980	4.5789	301.59	7238.23
97	9409	912673	9.8489	4.5947	304.73	7389.81
98	9604	941192	9.8995	4.6104	307.88	7542.96
99	9801	970299	9.9499	4.6261	311.02	7697.69
100	10000	1000000	10.0000	4.6416	314.16	7853.98
101	10201	1030301	10.0499	4.6570	317.30	8011.85
102	10404	1061208	10.0995	4.6723	320.44	8171.28
103	10609	1092727	10.1489	4.6875	323.58	8332.29
104	10816	1124864	10.1980	4.7027	326.73	8494.87
105	11025	1157625	10.2470	4.7177	329.87	8659.01
106	11236	1191016	10.2956	4.7326	333.01	8824.73
107	11449	1225043	10.3441	4.7475	336.15	8992.02
108	11664	1259712	10.3923	4.7622	339.29	9160.88
109	11881	1295029	10.4403	4.7769	342.43	9331.32
110	12100	1331000	10.4881	4.7914	345.58	9503.32
111	12321	1367631	10.5357	4.8059	348.72	9676.89
112	12544	1404928	10.5830	4.8203	351.86	9852.03
113	12769	1442897	10.6301	4.8346	355.00	10028.7
114	12996	1481544	10.6771	4.8488	358.14	10207.0
115	13225	1520875	10.7238	4.8629	361.28	10386.9
116	13456	1560896	10.7703	4.8770	364.42	10568.3
117	13689	1601613	10.8167	4.8910	367.57	10751.3
118	13924	1643032	10.8628	4.9049	370.71	10935.9
119	14161	1685159	10.9087	4.9187	373.85	11122.0
120	14400	1728000	10.9545	4.9324	376.99	11309.7

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
121	14641	1771561	11.0000	4.9461	380.13	11499.0
122	14884	1815848	11.0454	4.9597	383.27	11689.9
123	15129	1860867	11.0905	4.9732	386.42	11882.3
124	15376	1906624	11.1355	4.9866	389.56	12076.3
125	15625	1953125	11.1803	5.0000	392.70	12271.8
126	15876	2000376	11.2250	5.0133	395.84	12469.0
127	16129	2048383	11.2694	5.0265	398.98	12667.7
128	16384	2097152	11.3137	5.0397	402.12	12868.0
129	16641	2146689	11.3578	5.0528	405.27	13069.8
130	16900	2197000	11.4018	5.0658	408.41	13273.2
131	17161	2248091	11.4455	5.0788	411.55	13478.2
132	17424	2299968	11.4891	5.0916	414.69	13684.8
133	17689	2352637	11.5326	5.1045	417.83	13892.9
134	17956	2406104	11.5758	5.1172	420.97	14102.6
135	18225	2460375	11.6190	5.1299	424.12	14313.9
136	18496	2515456	11.6619	5.1426	427.26	14526.7
137	18769	2571353	11.7047	5.1551	430.40	14741.1
138	19044	2628072	11.7473	5.1676	433.54	14957.1
139	19321	2685619	11.7898	5.1801	436.68	15174.7
140	19600	2744000	11.8322	5.1925	439.82	15393.8
141	19881	2803221	11.8743	5.2048	442.96	15614.5
142	20164	2863288	11.9164	5.2171	446.11	15836.8
143	20449	2924207	11.9583	5.2293	449.25	16060.6
144	20736	2985984	12.0000	5.2415	452.39	16286.0
145	21025	3048625	12.0416	5.2536	455.53	16513.0
146	21316	3112136	12.0830	5.2656	458.67	16741.5
147	21609	3176523	12.1244	5.2776	461.81	16971.7
148	21904	3241792	12.1655	5.2896	464.96	17203.4
149	22201	3307949	12.2066	5.3015	468.10	17436.6
150	22500	3375000	12.2474	5.3133	471.24	17671.5
151	22801	3442951	12.2882	5.3251	474.38	17907.9
152	23104	3511808	12.3288	5.3368	477.52	18145.8
153	23409	3581577	12.3693	5.3485	480.66	18385.4
154	23716	3652264	12.4097	5.3601	483.81	18626.5
155	24025	3723875	12.4499	5.3717	486.95	18869.2
156	24336	3796416	12.4900	5.3832	490.09	19113.4
157	24649	3869893	12.5300	5.3947	493.23	19359.3
158	24964	3944312	12.5698	5.4061	496.37	19606.7
159	25281	4019679	12.6095	5.4175	499.51	19855.7
160	25600	4096000	12.6491	5.4288	502.65	20106.2

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
161	25921	4173281	12.6886	5.4401	505.80	20358.3
162	26244	4251528	12.7279	5.4514	508.94	20612.0
163	26569	4330747	12.7671	5.4626	512.08	20867.2
164	26896	4410944	12.8062	5.4737	515.22	21124.1
165	27225	4492125	12.8452	5.4848	518.36	21382.5
166	27556	4574296	12.8841	5.4959	521.50	21642.4
167	27889	4657463	12.9228	5.5069	524.65	21904.0
168	28224	4741632	12.9615	5.5178	527.79	22167.1
169	28561	4826809	13.0000	5.5288	530.93	22431.8
170	28900	4913000	13.0384	5.5397	534.07	22698.0
171	29241	5000211	13.0767	5.5505	537.21	22965.8
172	29584	5088448	13.1149	5.5613	540.35	23235.2
173	29929	5177717	13.1529	5.5721	543.50	23506.2
174	30276	5268024	13.1909	5.5828	546.64	23778.7
175	30625	5359375	13.2288	5.5934	549.78	24052.8
176	30976	5451776	13.2665	5.6041	552.92	24328.5
177	31329	5545233	13.3041	5.6147	556.06	24605.7
178	31684	5639752	13.3417	5.6252	559.20	24884.6
179	32041	5735339	13.3791	5.6357	562.35	25164.9
180	32400	5832000	13.4164	5.6462	565.49	25446.9
181	32761	5929741	13.4536	5.6567	568.63	25730.4
182	33124	6028568	13.4907	5.6671	571.77	26015.5
183	33489	6128487	13.5277	5.6774	574.91	26302.2
184	33856	6229504	13.5647	5.6877	578.05	26590.4
185	34225	6331625	13.6015	5.6980	581.19	26880.3
186	34596	6434856	13.6382	5.7083	584.34	27171.6
187	34969	6539203	13.6748	5.7185	587.48	27464.6
188	35344	6644672	13.7113	5.7287	590.62	27759.1
189	35721	6751269	13.7477	5.7388	593.76	28055.2
190	36100	6859000	15.7840	5.7489	596.90	28352.9
191	36481	6967871	13.8203	5.7590	600.04	28652.1
192	36864	7077888	13.8564	5.7690	603.19	28952.9
193	37249	7189057	13.8924	5.7790	606.33	29255.3
194	37636	7301384	13.9284	5.7890	609.47	29559.2
195	38025	7414875	13.9642	5.7989	612.61	29864.8
196	38416	7529536	14.0000	5.8088	615.75	30171.9
197	38809	7645373	14.0357	5.8186	618.89	30480.5
198	39204	7762392	14.0712	5.8285	622.04	30790.7
199	39601	7880599	14.1067	5.8383	625.18	31102.6
200	40000	8000000	14.1421	5.8480	628.32	31415.9

No.	Square	Cube	Sq. Root	Cube Root	Circle Circum.	Circle Area
201	40401	8120601	14.1774	5.8578	631.46	31730.9
202	40804	8242408	14.2127	5.8675	634.60	32047.4
203	41209	8365427	14.2478	5.8771	637.74	32365.5
204	41616	8489664	14.2829	5.8868	640.89	32685.1
205	42025	8615125	14.3178	5.8964	644.03	33006.4
206	42436	8741816	14.3527	5.9059	647.17	33329.2
207	42849	8869743	14.3875	5.9155	650.31	33653.5
208	43264	8998912	14.4222	5.9250	653.45	33979.5
209	43681	9129329	14.4568	5.9345	656.59	34307.0
210	44100	9261000	14.4914	5.9439	659.73	34636.1
211	44521	9393931	14.5258	5.9533	662.88	34966.7
212	44944	9528128	14.5602	5.9627	666.02	35298.9
213	45369	9663597	14.5945	5.9721	669.16	35632.7
214	45796	9800344	14.6287	5.9814	672.30	35968.1
215	46225	9938375	14.6629	5.9907	675.44	36305.0
216	46656	10077696	14.6969	6.0000	678.58	36643.5
217	47089	10218313	14.7309	6.0092	681.73	36983.6
218	47524	10360232	14.7648	6.0185	684.87	37325.3
219	47961	10503459	14.7986	6.0277	688.01	37668.5
220	48400	10648000	14.8324	6.0368	691.15	38013.3
221	48841	10793861	14.8661	6.0459	694.29	38359.6
222	49284	10941048	14.8997	6.0550	697.43	38707.6
223	49729	11089567	14.9332	6.0641	700.58	39057.1
224	50176	11239424	14.9666	6.0732	703.72	39408.1
225	50625	11390625	15.0000	6.0822	706.86	39760.8
226	51076	11543176	15.0333	6.0912	710.00	40115.0
227	51529	11697083	15.0665	6.1002	713.14	40470.8
228	51984	11852352	15.0997	6.1091	716.28	40828.1
229	52441	12008989	15.1327	6.1180	719.42	41187.1
230	52900	12167000	15.1658	6.1269	722.57	41547.6
231	53361	12326391	15.1987	6.1358	725.71	41909.6
232	53824	12487168	15.2315	6.1446	728.85	42273.3
233	54289	12649337	15.2643	6.1534	731.99	42638.5
234	54756	12812904	15.2971	6.1622	735.13	43005.3
235	55225	12977875	15.3297	6.1710	738.27	43373.6
236	55696	13144256	15.3623	6.1797	741.42	43743.5
237	56169	13312053	15.3948	6.1885	744.56	44115.0
238	56644	13481272	15.4272	6.1972	747.70	44488.1
239	57121	13651919	15.4596	6.2058	750.84	44862.7
240	57600	13824000	15.4919	6.2145	753.98	45238.9

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
241	58081	13997521	15.5242	6.2231	757.12	45616.7
242	58564	14172488	15.5563	6.2317	760.27	45996.1
243	59049	14348907	15.5885	6.2403	763.41	46377.0
244	59536	14526784	15.6205	6.2488	766.55	46759.5
245	60025	14706125	15.6525	6.2573	769.69	47143.5
246	60516	14886936	15.6844	6.2658	772.83	47529.2
247	61009	15069223	15.7162	6.2743	775.97	47916.4
248	61504	15252992	15.7480	6.2828	779.12	48305.1
249	62001	15438249	15.7797	6.2912	782.26	48695.5
250	62500	15625000	15,8114	6.2996	785.40	49087.4
251	63001	15813251	15.8430	6.3080	788.54	49480.9
252	63504	16003008	15.8745	6.3164	791.68	49875.9
253	64009	16194277	15.9060	6.3247	794.82	50272.6
254	64516	16387064	15.9374	6.3330	797.96	50670.7
255	65025	16581375	15.9687	6.3413	801.11	51070.5
256	65536	16777216	16.0000	6.3496	804.25	51471.9
257	66049	16974593	16.0312	6.3579	807.39	51874.8
258	66564	17173512	16.0624	6.3661	810.53	52279.2
259	67081	17373979	16.0935	6.3743	813.67	52685.3
260	67600	17576000	16.1245	6.3825	816.81	53092.9
261	68121	17779581	16.1555	6.3907	819.96	53502.1
262	68644	17984728	16.1864	6.3988	823.10	53912.9
263	69169	18191447	16.2173	6.4070	826.24	54325.2
264	69696	18399744	16.2481	6.4151	829.38	54739.1
265	70225	18609625	16.2788	6.4232	832.52	55154.6
266	70756	18821096	16.3095	6.4312	835.66	55571.6
267	71289	19034163	16.3401	6.4393	838.81	55990.3
268	71824	19248832	16.3707	6.4473	841.95	56410.4
269	72361	19465109	16.4012	6.4553	845.09	56832.2
270	72900	19683000	16.4317	6.4633	848.23	57255.5
271	73441	19902511	16.4621	6.4713	851.37	57680.4
272	73984	20123648	16.4924	6.4792	854.51	58106.9
273	74529	20346417	16.5227	6.4872	857.66	58534.9
274	75076	20570824	16.5529	6.4951	860.80	58964.6
275	75625	20796875	16.5831	6.5030	863.94	59395.7
276	76176	21024576	16.6132	6.5108	867.08	59828.5
277	76729	21253933	16.6433	6.5187	870.22	60262.8
278	77284	21484952	16.6733	6.5265	873.36	60698.7
279	77841	21717639	16.7033	6.5343	876.50	61136.2
280	78400	21952000	16.7332	6.5421	879.65	61575.2

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
281	78961	22188041	16.7631	6.5499	882.79	62015.8
282	79524	22425768	16.7929	6.5577	885.93	62458.0
283	80089	22665187	16.8226	6.5654	889.07	62901.8
284	80656	22906304	16.8523	6.5731	892.21	63347.1
285	81225	23149125	16.8819	6.5808	895.35	63794.0
286	81796	23393656	16.9115	6.5885	898.50	64242.4
287	82369	23639903	16.9411	6.5962	901.64	64692.5
288	82944	23887872	16.9706	6.6039	904.78	65144.1
289	83521	24137569	17.0000	6.6115	907.92	65597.2
290	84100	24389000	17.0294	6.6191	911.06	66052.0
291	84681	24642171	17.0587	6.6267	914.20	66508.3
292	85264	24897088	17.0880	6.6343	917.35	66966.2
293	85849	25153757	17.1172	6.6419	920.49	67425.6
294	86436	25412184	17.1464	6.6494	923.63	67886.7
295	87025	25672375	17.1756	6.6569	926.77	68349.3
296	87616	25934336	17.2047	6.6644	929.91	68813.5
297	88209	26198073	17.2337	6.6719	933.05	69279.2
298	88804	26463592	17.2627	6.6794	936.19	69746.5
299	89401	26730899	17.2916	6.6869	939.34	70215.4
300	90000	27000000	17.3205	6.6943	942.48	70685.8
301	90601	27270901	17.3494	6.7018	945.62	71157.9
302	91204	27543608	17.3781	6.7092	948.76	71631.5
303	91809	27818127	17.4069	6.7166	951.90	72106.6
304	92416	28094464	17.4356	6.7240	955.04	72583.4
305	93025	28372625	17.4642	6.7313	958.19	73061.7
306	93636	28652616	17.4929	6.7387	961.33	73541.5
307	94249	28934443	17.5214	6.7460	964.47	74023.0
308	94864	29218112	17.5499	6.7533	967.61	74506.0
309	95481	29503629	17.5784	6.7606	970.75	74990.6
310	96100	29791000	17.6068	6.7679	973.89	75476.8
311	96721	30080231	17.6352	6.7752	977.04	75964.5
312	97344	30371328	17.6635	6.7824	980.18	76453.8
313	97969	30664297	17.6918	6.7897	983.32	76944.7
314	98596	30959144	17.7200	6.7969	986.46	77437.1
315	99225	31255875	17.7482	6.8041	989.60	77931.1
316	99856	31554496	17.7764	6.8113	992.74	78426.7
317	100489	31855013	17.8045	6.8185	995.88	78923.9
318	101124	32157432	17.8326	6.8256	999.03	79422.6
319	101761	32461759	17.8606	6.8328	1002.20	79922.9
320	102400	32768000	17.8885	6.8399	1005.30	80424.8

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
321	103041	33076161	17.9165	6.8470	1008.5	80928.2
322	103684	33386248	17.9444	6.8541	1011.6	81433.2
323	104329	33698267	17.9722	6.8612	1014.7	81939.8
324	104976	34012224	18.0000	6.8683	1017.9	82448.0
325	105625	34328125	18.0278	6.8753	1021.0	82957.7
326	106276	34645976	18.0555	6.8824	1024.2	83469.0
327	106929	34965783	18.0831	6.8894	1027.3	83981.8
328	107584	35287552	18.1108	6.8964	1030.4	84496.3
329	108241	35611289	18.1384	6.9034	1033.6	85012.3
330	108900	35937000	18.1659	6.9104	1036.7	85529.9
331	109561	36264691	18.1934	6.9174	1039.9	86049.0
332	110224	36594368	18.2209	6.9244	1043.0	86569.7
333	110889	36926037	18.2483	6.9313	1046.2	87092.0
334	111556	37259704	18.2757	6.9382	1049.3	87615.9
335	112225	37595375	18.3030	6.9451	1052.4	88141.3
336	112896	37933056	18.3303	6.9521	1055.6	88668.3
337	113569	38272753	18.3576	6.9589	1058.7	89196.9
338	114244	38614472	18.3848	6.9658	1061.9	89727.0
339	114921	38958219	18.4120	6.9727	1065.0	90258.7
340	115600	39304000	18.4391	6.9795	1068.1	90792.0
341	116281	39651821	18.4662	6.9864	1071.3	91326.9
342	116964	40001688	18.4932	6.9932	1074.4	91863.3
343	117649	40353607	18.5203	7.0000	1077.6	92401.3
344	118336	40707584	18.5472	7.0068	1080.7	92940.9
345	119025	41063625	18.5742	7.0136	1083.8	93482.0
346	119716	41421736	18.6011	7.0203	1087.0	94024.7
347	120409	41781923	18.6279	7.0271	1090.1	94569.0
348	121104	42144192	18.6548	7.0338	1093.3	95114.9
349	121801	42508549	18.6815	7.0406	1096.4	95662.3
350	122500	42875000	18.7083	7.0473	1099.6	96211.3
351	123201	43243551	18.7350	7.0540	1102.7	96761.8
352	123904	43614208	18.7617	7.0607	1105.8	97314.0
353	124609	43986977	18.7883	7.0674	1109.0	97867.7
354	125316	44361864	18.8149	7.0740	1112.1	98423.0
355	126025	44738875	18.8414	7.0807	1115.3	98979.8
356	126736	45118016	18.8680	7.0873	1118.4	99538.2
357	127449	45499293	18.8944	7.0940	1121.5	100098
358	128164	45882712	18.9209	7.1006	1124.7	100660
359	128881	46268279	18.9473	7.1072	1127.8	101223
360	129600	46656000	18.9737	7.1138	1131.0	101788

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
361	130321	47045881	19.0000	7.1204	1134.1	102354
362	131044	47437928	19.0263	7.1269	1137.3	102922
363	131769	47832147	19.0526	7.1335	1140.4	103491
364	132496	48228544	19.0788	7.1400	1143.5	104062
365	133225	48627125	19.1050	7.1466	1146.7	104635
366	133956	49027896	19.1311	7.1531	1149.8	105209
367	134689	49430863	19.1572	7.1596	1153.0	105785
368	135424	49836032	19.1833	7.1661	1156.1	106362
369	136161	50243409	19.2094	7.1726	1159.2	106941
370	136900	50653000	19.2354	7.1791	1162.4	107521
371	137641	51064811	19.2614	7.1855	1165.5	108103
372	138384	51478848	19.2873	7.1920	1168.7	108687
373	139129	51895117	19.3132	7.1984	1171.8	109272
374	139876	52313624	19.3391	7.2048	1175.0	109858
375	140625	52734375	19.3649	7.2112	1178.1	110447
376	141376	53157376	19.3907	7.2177	1181.2	111036
377	142129	53582633	19.4165	7.2240	1184.4	111628
378	142884	54010152	19.4422	7.2304	1187.5	112221
379	143641	54439939	19.4679	7.2368	1190.7	112815
380	144400	54872000	19.4936	7.2432	1193.8	113411
381	145161	55306341	19.5192	7.2495	1196.9	114009
382	145924	55742968	19.5448	7.2558	1200.1	114608
383	146689	56181887	19.5704	7.2622	1203.2	115209
384	147456	56623104	19.5959	7.2685	1206.4	115812
385	148225	57066625	19.6214	7.2748	1209.5	116416
386	148996	57512456	19.6469	7.2811	1212.7	117021
387	149769	57960603	19.6723	7.2874	1215.8	117628
388	150544	58411072	19.6977	7.2936	1218.9	118237
389	151321	58863869	19.7231	7.2999	1222.1	118847
390	152100	59319000	19.7484	7.3061	1225.2	119459
391	152881	59776471	19.7737	7.3124	1228.4	120072
392	153664	60236288	19.7990	7.3186	1231.5	120687
393	154449	60698457	19.8242	7.3248	1234.6	121304
394	155236	61162984	19.8494	7.3310	1237.8	121922
395	156025	61629875	19.8746	7.3372	1240.9	122542
396	156816	62099136	19.8997	7.3434	1244.1	123163
397	157609	62570773	19.9249	7.3496	1247.2	123786
398	158404	63044792	19.9499	7.3558	1250.4	124410
399	159201	63521199	19.9750	7.3619	1253.5	125036
400	160000	64000000	20.0000	7.3684	1256.6	125664

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES, AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
401	160801	64481201	20.0250	7.3742	1259.8	126293
402	161604	64964808	20.0499	7.3803	1262.9	126923
403	162409	65450827	20.0749	7.3864	1266.1	127556
404	163216	65939264	20.0998	7.3925	1269.2	128190
405	164025	66430125	20.1246	7.3986	1272.3	128825
406	164836	66923416	20.1494	7.4047	1275.5	129462
407	165649	67419143	20.1742	7.4108	1278.6	130100
408	166464	67917312	20.1990	7.4169	1281.8	130741
409	167281	68417929	20.2237	7.4229	1284.9	131382
410	168100	68921000	20.2485	7.4290	1288.1	132025
411	168921	69426531	20.2731	7.4350	1291.2	132670
412	169744	69934528	20.2978	7.4410	1294.3	133317
413	170569	70444997	20.3224	7.4470	1297.5	133965
414	171396	70957944	20.3470	7.4530	1300.6	134614
415	172225	71473375	20.3715	7.4590	1303.8	135265
416	173056	71991296	20.3961	7.4650	1306.9	135918
417	173889	72511713	20.4206	7.4710	1310.0	136572
418	174724	73034632	20.4450	7.4770	1313.2	137228
419	175561	73560059	20.4695	7.4829	1316.3	137885
420	176400	74088000	20.4939	7.4889	1319.5	138544
421	177241	74618461	20.5183	7.4948	1322.6	139205
422	178084	75151448	20.5426	7.5007	1325.8	139867
423	178929	75686967	20.5670	7.5067	1328.9	140531
424	179776	76225024	20.5913	7.5126	1332.0	141196
425	180625	76765625	20.6155	7.5185	1335.2	141863
426	181476	77308776	20.6398	7.5244	1338.3	142531
427	182329	77854483	20.6640	7.5302	1341.5	143201
428	183184	78402752	20.6882	7.5361	1344.6	143872
429	184041	78953589	20.7123	7.5420	1347.7	144545
430	184900	79507000	20.7364	7.5478	1350.9	145220
431	185761	80062991	20.7605	7.5537	1354.0	145896
432	186624	80621568	20.7846	7.5595	1357.2	146574
433	187489	81182737	20.8087	7.5654	1360.3	147254
434	188356	81746504	20.8327	7.5712	1363.5	147934
435	189225	82312875	20.8567	7.5770	1366.6	148617
436	190096	82881856	20.8806	7.5828	1369.7	149301
437	190969	83453453	20.9045	7.5886	1372.9	149987
438	191844	84027672	20.9284	7.5944	1376.0	150674
439	192721	84604519	20.9523	7.6001	1379.2	151363
440	193600	85184000	20.9762	7.6059	1382.3	152053

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circum.	Area
441	194481	85766121	21.0000	7.6117	1385.4	152745
442	195364	86350888	21.0238	7.6174	1388.6	153439
443	196249	86938307	21.0476	7.6232	1391.7	154134
444	197136	87528384	21.0713	7.6289	1394.9	154830
445	198025	88121125	21.0950	7.6346	1398.0	155528
446	198916	88716536	21.1187	7.6403	1401.2	156228
447	199809	89314623	21.1424	7.6460	1404.3	156930
448	200704	89915392	21.1660	7.6517	1407.4	157633
449	201601	90518849	21.1896	7.6574	1410.6	158337
450	202500	91125000	21.2132	7.6631	1413.7	159043
451	203401	91733851	21.2368	7.6688	1416.9	159751
452	204304	92345408	21.2603	7.6744	1420.0	160460
453	205209	92959677	21.2838	7.6801	1423.1	161171
454	206116	93576664	21.3073	7.6857	1426.3	161883
455	207025	94196375	21.3307	7.6914	1429.4	162597
456	207936	94818816	21.3542	7.6970	1432.6	163313
457	208849	95443993	21.3776	7.7026	1435.7	164030
458	209764	96071912	21.4009	7.7082	1438.9	164748
459	210681	96702579	21.4243	7.7138	1442.0	165468
460	211600	97336000	21.4476	7.7194	1445.1	166190
461	212521	97972181	21.4709	7.7250	1448.3	166914
462	213444	98611128	21.4942	7.7306	1451.4	167639
463	214369	99252847	21.5174	7.7362	1454.6	168365
464	215296	99897344	21.5407	7.7418	1457.7	169093
465	216225	100544625	21.5639	7.7473	1460.8	169823
466	217156	101194696	21.5870	7.7529	1464.0	170554
467	218089	101847563	21.6102	7.7584	1467.1	171287
468	219024	102503232	21.6333	7.7639	1470.3	172021
469	219961	103161709	21.6564	7.7695	1473.4	172757
470	220900	103823000	21.6795	7.7750	1476.5	173494
471	221841	104487111	21.7025	7.7805	1479.7	174234
472	222784	105154048	21.7256	7.7860	1482.8	174974
473	223729	105823817	21.7486	7.7915	1486.0	175716
474	224676	106496424	21.7715	7.7970	1489.1	176460
475	225625	107171875	21.7945	7.8025	1492.3	177205
476	226576	107850176	21.8174	7.8079	1495.4	177952
477	227529	108531333	21.8403	7.8134	1498.5	178701
478	228484	109215352	21.8632	7.8188	1501.7	179451
479	229441	109902239	21.8861	7.8243	1504.8	180203
480	230400	110592000	21.9089	7.8297	1508.0	180956

TABLE III.—SQUARES, CUBES, SQUARE ROOTS, CUBE ROOTS, CIRCUMFERENCES AND CIRCULAR AREAS OF NOS. FROM 1 TO 520 (*Continued*)

No.	Square	Cube	Sq. Root	Cube Root	Circle Circum.	Circle Area
481	231361	111284641	21.9317	7.8352	1511.1	181711
482	232324	111980168	21.9545	7.8406	1514.3	182467
483	233289	112678587	21.9773	7.8460	1517.4	183225
484	234256	113379904	22.0000	7.8514	1520.5	183984
485	235225	114084125	22.0227	7.8568	1523.7	184745
486	236196	114791256	22.0454	7.8622	1526.8	185508
487	237169	115501303	22.0681	7.8676	1530.0	186272
488	238144	116214272	22.0907	7.8730	1533.1	187038
489	239121	116930169	22.1133	7.8784	1536.2	187805
490	240100	117649000	22.1359	7.8837	1539.4	188574
491	241081	118370771	22.1585	7.8891	1542.5	189345
492	242064	119095488	22.1811	7.8944	1545.7	190117
493	243049	119823157	22.2036	7.8998	1548.8	190890
494	244036	120553784	22.2261	7.9051	1551.9	191665
495	245025	121287375	22.2486	7.9105	1555.1	192442
496	246016	122023936	22.2711	7.9158	1558.2	193221
497	247009	122763473	22.2935	7.9211	1561.4	194000
498	248004	123505992	22.3159	7.9264	1564.5	194782
499	249001	124251499	22.3383	7.9317	1567.7	195565
500	250000	125000000	22.3607	7.9370	1570.8	196350
501	251001	125751501	22.3830	7.9423	1573.9	197136
502	252004	126506008	22.4054	7.9476	1577.1	197923
503	253009	127263527	22.4277	7.9528	1580.2	198713
504	254016	128024064	22.4499	7.9581	1583.4	199504
505	255025	128787625	22.4722	7.9634	1586.5	200296
506	256036	129554216	22.4944	7.9686	1589.7	201090
507	257049	130323843	22.5167	7.9739	1592.8	201886
508	258064	131096512	22.5389	7.9791	1595.9	202683
509	259081	131872229	22.5610	7.9843	1599.1	203482
510	260100	132651000	22.5832	7.9896	1602.2	204282
511	261121	133432831	22.6053	7.9948	1605.4	205084
512	262144	134217728	22.6274	8.0000	1608.5	205887
513	263169	135005697	22.6495	8.0052	1611.6	206692
514	264196	135796744	22.6716	8.0104	1614.8	207499
515	265225	136590875	22.6936	8.0156	1617.9	208307
516	266256	137388096	22.7156	8.0208	1621.1	209117
517	267289	138188413	22.7376	8.0260	1624.2	209928
518	268324	138991832	22.7596	8.0311	1627.3	210741
519	269361	139798359	22.7816	8.0363	1630.5	211556
520	270400	140608000	22.8035	8.0415	1633.6	212372

TABLE IV.—FACTORS OF EVAPORATION

Temperature of feed-water	Boiler gage pressures in pounds per square inch above the atmosphere									
	0	5	10	15	20	25	30	35	40	45
° Fahr.										
32	1.187	1.192	1.195	1.199	1.201	1.204	1.206	1.209	1.211	1.212
35	1.184	1.189	1.192	1.196	1.198	1.201	1.203	1.206	1.208	1.209
40	1.179	1.184	1.187	1.191	1.193	1.196	1.198	1.201	1.203	1.204
45	1.173	1.178	1.181	1.185	1.187	1.190	1.192	1.195	1.197	1.198
50	1.168	1.173	1.177	1.180	1.182	1.185	1.187	1.190	1.192	1.193
55	1.163	1.168	1.171	1.175	1.177	1.180	1.182	1.185	1.187	1.188
60	1.158	1.163	1.166	1.170	1.172	1.175	1.177	1.180	1.182	1.183
65	1.153	1.158	1.161	1.165	1.167	1.170	1.172	1.175	1.177	1.178
70	1.148	1.153	1.156	1.160	1.162	1.165	1.167	1.170	1.172	1.173
75	1.143	1.148	1.151	1.155	1.157	1.160	1.162	1.165	1.167	1.168
80	1.137	1.143	1.146	1.149	1.151	1.154	1.156	1.159	1.161	1.162
85	1.132	1.137	1.140	1.144	1.146	1.149	1.151	1.154	1.156	1.157
90	1.127	1.132	1.135	1.139	1.141	1.144	1.146	1.149	1.151	1.152
95	1.122	1.127	1.130	1.134	1.136	1.139	1.141	1.144	1.146	1.147
100	1.117	1.122	1.125	1.129	1.131	1.134	1.136	1.139	1.141	1.142
105	1.111	1.117	1.120	1.123	1.125	1.128	1.130	1.133	1.135	1.136
110	1.106	1.111	1.114	1.118	1.120	1.123	1.125	1.128	1.130	1.131
115	1.101	1.106	1.109	1.113	1.115	1.118	1.120	1.123	1.125	1.126
120	1.096	1.101	1.104	1.108	1.110	1.113	1.115	1.118	1.120	1.121
125	1.091	1.096	1.099	1.103	1.105	1.108	1.110	1.113	1.115	1.116
130	1.085	1.091	1.094	1.097	1.099	1.102	1.104	1.107	1.109	1.110
135	1.080	1.085	1.088	1.092	1.094	1.097	1.099	1.102	1.104	1.105
140	1.075	1.080	1.083	1.087	1.089	1.092	1.094	1.097	1.099	1.100
145	1.070	1.075	1.078	1.082	1.084	1.087	1.089	1.092	1.094	1.095
150	1.065	1.070	1.073	1.077	1.079	1.082	1.084	1.087	1.089	1.090
155	1.059	1.065	1.068	1.071	1.073	1.076	1.078	1.081	1.083	1.084
160	1.054	1.059	1.062	1.066	1.068	1.071	1.073	1.076	1.078	1.079
165	1.049	1.054	1.057	1.061	1.063	1.066	1.068	1.071	1.073	1.074
170	1.044	1.049	1.052	1.056	1.058	1.061	1.063	1.066	1.068	1.069
175	1.039	1.044	1.047	1.051	1.053	1.056	1.058	1.061	1.063	1.064
180	1.033	1.039	1.042	1.045	1.047	1.050	1.052	1.055	1.057	1.058
185	1.028	1.033	1.036	1.040	1.042	1.045	1.047	1.050	1.052	1.053
190	1.023	1.028	1.031	1.035	1.037	1.040	1.042	1.045	1.047	1.048
195	1.018	1.023	1.025	1.030	1.032	1.035	1.037	1.040	1.042	1.043
200	1.013	1.018	1.021	1.025	1.027	1.030	1.032	1.035	1.037	1.038
205	1.008	1.013	1.015	1.020	1.022	1.025	1.027	1.030	1.032	1.033
210	1.008	1.008	1.011	1.015	1.017	1.020	1.022	1.025	1.027	1.028
212	1.002	1.002	.....	.....	.....	.....	.....	.....	.....	.....

TABLE IV.—FACTORS OF EVAPORATION (*Continued*)

Temperature of feed-water	Boiler gage pressures in pounds per square inch above the atmosphere									
	50	60	70	80	90	100	120	140	160	180
° Fahr.										
32	1.214	1.217	1.219	1.222	1.224	1.227	1.231	1.234	1.237	1.239
35	1.211	1.214	1.216	1.219	1.221	1.224	1.228	1.231	1.234	1.236
40	1.206	1.209	1.211	1.214	1.216	1.219	1.223	1.226	1.229	1.231
45	1.200	1.203	1.205	1.208	1.210	1.213	1.217	1.220	1.223	1.225
50	1.195	1.198	1.200	1.203	1.205	1.208	1.212	1.215	1.218	1.220
55	1.190	1.193	1.195	1.198	1.200	1.203	1.207	1.210	1.213	1.215
60	1.185	1.188	1.190	1.193	1.195	1.198	1.202	1.205	1.208	1.210
65	1.180	1.183	1.185	1.188	1.190	1.193	1.197	1.200	1.203	1.205
70	1.175	1.178	1.180	1.183	1.185	1.188	1.192	1.195	1.198	1.200
75	1.170	1.173	1.175	1.178	1.180	1.183	1.187	1.190	1.193	1.195
80	1.164	1.167	1.169	1.172	1.174	1.177	1.181	1.184	1.187	1.189
85	1.159	1.162	1.164	1.167	1.169	1.172	1.176	1.179	1.182	1.184
90	1.154	1.157	1.159	1.162	1.164	1.167	1.171	1.174	1.177	1.179
95	1.149	1.152	1.154	1.157	1.159	1.162	1.166	1.169	1.172	1.174
100	1.144	1.147	1.149	1.152	1.154	1.157	1.161	1.164	1.167	1.169
105	1.138	1.141	1.143	1.146	1.148	1.151	1.155	1.158	1.161	1.163
110	1.133	1.136	1.138	1.141	1.143	1.146	1.150	1.153	1.156	1.158
115	1.128	1.131	1.133	1.136	1.138	1.141	1.145	1.148	1.151	1.153
120	1.123	1.126	1.128	1.131	1.133	1.136	1.140	1.143	1.146	1.148
125	1.118	1.121	1.123	1.126	1.128	1.131	1.135	1.138	1.141	1.143
130	1.112	1.115	1.117	1.120	1.122	1.125	1.129	1.132	1.135	1.137
135	1.107	1.110	1.112	1.115	1.117	1.120	1.124	1.127	1.130	1.132
140	1.102	1.105	1.107	1.110	1.112	1.115	1.119	1.122	1.125	1.127
145	1.097	1.100	1.102	1.105	1.107	1.110	1.114	1.117	1.120	1.122
150	1.092	1.095	1.097	1.100	1.102	1.105	1.109	1.112	1.115	1.117
155	1.086	1.089	1.091	1.094	1.096	1.099	1.103	1.106	1.109	1.111
160	1.081	1.084	1.086	1.089	1.091	1.094	1.098	1.101	1.104	1.106
165	1.076	1.079	1.081	1.084	1.086	1.089	1.093	1.096	1.099	1.101
170	1.071	1.074	1.076	1.079	1.081	1.084	1.088	1.091	1.094	1.096
175	1.066	1.069	1.071	1.074	1.076	1.079	1.083	1.086	1.089	1.091
180	1.060	1.063	1.065	1.068	1.070	1.073	1.077	1.080	1.083	1.085
185	1.055	1.058	1.060	1.063	1.065	1.068	1.072	1.075	1.078	1.080
190	1.050	1.053	1.055	1.058	1.060	1.063	1.067	1.070	1.073	1.075
195	1.045	1.048	1.050	1.053	1.055	1.058	1.062	1.065	1.068	1.070
200	1.040	1.043	1.045	1.048	1.050	1.053	1.057	1.060	1.063	1.065
205	1.035	1.038	1.040	1.043	1.045	1.048	1.052	1.055	1.058	1.060
210	1.030	1.033	1.035	1.038	1.040	1.043	1.047	1.050	1.053	1.055
212	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....

How to Interpolate the Table of Factors of Evaporation

It sometimes happens when it is desired to use the table of factors of evaporation that the given figure for any case falls between two certain figures in the table, and therefore the correct result cannot at once be found without resorting to what is called "interpolation."

Suppose, for example, that the average steam pressure in the boiler is 64 lb. per square inch gage, and that the average temperature of the feed water is 132° F.; what is the factor of evaporation? By referring to the table, there are no columns with heading or side heading corresponding to these figures, and unless there is some definite method of obtaining exact figures, it would be necessary to strike an average between two sets of figures in the table, nearest to those given in the example. While for ordinary purposes this would be close enough, yet because of the ease with which the real figures may be found it is worth while to learn what to do.

The factor for 60 lb. gage and 130° feed water is 1.115; the factor for 70 lb. gage and 130° feed water is 1.118; the factor for 64 lb. gage amd 130° feed water is therefore,

$$1.115 + \frac{1.118 - 1.115}{10} \times 4 = 1.1162$$

In the same manner, the factor of evaporation for 64 lb. gage pressure and 140° feed water is found to be

$$1.105 + \frac{1.107 - 1.105}{10} \times 4 = 1.1058$$

There is now the factor for 64 lb. gage and 130° feed water, and 64 lb. gage and 140° feed water, and it only remains to interpolate between these values to get the factor for 64 lb. gage and 132° feed water. This is done as follows:

$$1.1162 - \frac{1.1162 - 1.1058}{10} \times 2 = 1.1141$$

which is the factor of evaporation corresponding to 64 lb. gage pressure and 132° F. feed water, as given in the example.

The foregoing method may be applied to any figures within the range of the table.

TABLE V.—STANDARD BOILER TUBES
Table of Standard Dimensions

Diameter		Standard thickness		Transverse areas		Area of surface per foot of tube		Nominal weight per foot-lb.				
External	Internal	Nearest B.W.G.	Inches	External	Internal	External	Internal	Standard thickness	One extra wire gage	Two extra wire gages	Three extra wire gages	Four extra wire gages
In.	In.	No.		sq. in.	sq. in.	sq. in.	sq. in.					
1	.810	13	.095	.785	.515	.262	.212	.90	1.04	1.13	1.24	1.35
1 1/4	1.060	13	.095	1.227	.882	.327	.277	1.15	1.33	1.45	1.60	1.74
1 1/2	1.310	13	.095	1.767	1.348	.392	.343	1.40	1.62	1.77	1.96	2.14
1 3/4	1.560	13	.095	2.405	1.911	.458	.408	1.66	1.91	2.09	2.31	2.53
2	1.810	13	.095	3.142	2.573	.523	.474	1.91	2.20	2.41	2.67	2.93
2 1/4	2.060	13	.095	3.976	3.333	.589	.539	2.16	2.49	2.73	3.03	3.32
2 1/2	2.282	12	.109	4.909	4.090	.654	.597	2.75	3.05	3.39	3.72	4.12
2 3/4	2.532	12	.109	5.940	5.035	.720	.663	3.04	3.37	3.74	4.11	4.56
3	2.782	12	.109	7.069	6.079	.785	.728	3.33	3.69	4.10	4.51	5.00
3 1/4	3.010	11	.120	8.296	7.116	.851	.788	3.96	4.46	4.90	5.44	5.90
3 1/2	3.260	11	.120	9.621	8.347	.916	.853	4.28	4.82	5.30	5.88	6.38
3 3/4	3.510	11	.120	11.045	9.676	.982	.919	4.60	5.18	5.69	6.32	6.86
4	3.732	10	.134	12.566	10.939	1.047	.977	5.47	6.09	6.76	7.34	8.23
4 1/2	4.232	10	.134	15.904	14.066	1.178	1.108	6.17	6.88	7.64	8.31	9.32
5	4.704	9	.148	19.635	17.379	1.309	1.231	7.58	8.52	9.27	10.40	11.23
6	5.670	8	.165	28.274	25.250	1.571	1.484	10.16	11.19	12.57	13.58	14.65

TABLE VI.—KENT'S TABLE OF

Formula: $H.\ P. = 3.33(A - 0.6\sqrt{A})\sqrt{H}$.

Diameter, inches	Area A sq. feet	Effective area, $E = A - 0.6\sqrt{A}$ sq. feet	Height of chimney					
			50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.
			Commercial horse-power of boiler					
18	1.77	.97	23	25	27	29	......	......
21	2.41	1.47	35	38	41	44	......	......
24	3.14	2.08	49	54	58	62	66	......
27	3.98	2.78	65	72	78	83	88	......
30	4.91	3.58	84	92	100	107	113	119
33	5.94	4.48	......	115	125	133	141	149
36	7.07	5.47	......	141	152	163	173	182
39	8.30	6.57	......	......	183	196	208	219
42	9.62	7.76	......	......	216	231	245	258
48	12.57	10.44	......	......	......	311	330	348
54	15.90	13.51	......	......	......	......	427	449
60	19.64	16.98	......	......	......	......	536	565
66	23.76	20.83	......	......	......	......	......	694
72	28.27	25.08	......	......	......	......	......	835
78	33.18	29.73	......	......	......	......	......	......
84	33.48	34.76	......	......	......	......	......	......
90	44.18	40.19	......	......	......	......	......	......
96	50.27	46.01	......	......	......	......	......	......
102	56.75	52.23	......	......	......	......	......	......
108	63.62	58.83	......	......	......	......	......	......
114	70.88	65.83	......	......	......	......	......	......
120	78.54	73.22	......	......	......	......	......	......
132	95.03	89.18	......	......	......	......	......	......
144	113.10	106.72	......	......	......	......	......	......

For pounds of coal burned per hour for any given size of chimney,

SIZE OF CHIMNEYS FOR STEAM BOILERS

(Assuming 1 H.P. = 5 lb. of coal burned per hour)

Height of chimney								Equivalent square chimney; side of square, $\sqrt{E}+4$ in.
110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	225 ft.	250 ft.	300 ft.	
Commercial horse-power of boiler								
...	...	...	...	...	...	...	...	16
...	...	...	...	...	...	...	...	19
...	...	...	...	...	...	...	...	22
...	...	...	...	...	...	...	...	24
...	...	...	...	...	...	...	...	27
156	...	...	...	...	...	...	...	30
191	204	...	...	...	...	...	...	32
229	245	...	...	...	...	...	...	35
271	289	316	...	...	...	...	...	38
365	389	426	...	...	...	...	...	43
472	503	551	595	...	...	...	...	48
593	632	692	748	...	...	...	...	54
728	776	849	918	981	...	...	...	59
876	934	1023	1105	1181	1253	...	...	64
1038	1107	1212	1310	1400	1485	1565	...	70
1214	1294	1418	1531	1637	1736	1830	2005	75
...	1496	1639	1770	1893	2008	2116	2318	80
...	1712	1876	2027	2167	2298	2423	2654	86
...	1944	2130	2300	2459	2609	2750	3012	91
...	2090	2399	2592	2771	2939	3098	3393	96
...	...	2685	2900	3100	3288	3466	3797	101
...	...	2986	3226	3448	3657	3855	4223	107
...	...	3637	3929	4200	4455	4696	5144	117
...	...	4352	4701	5026	5331	5618	6155	128

multiply the figures in the table by 5.

TABLE VII.—PROPERTIES OF SATURATED STEAM[1]

1 Pressure lb. per sq. in. P	2 Temp. degrees F	3 Vol. cu. ft. per lb. V or S	4 Weight, lb. per cu. ft. V	5 Heat of the liquid Q	6 Latent heat of evap. L or R	7 Total heat of steam H
1	101.83	333.0	0.00300	69.8	1034.6	1104.4
2	126.15	173.5	0.00576	94.0	1021.0	1115.0
3	141.52	118.5	0.00845	109.4	1012.3	1121.6
4	153.01	90.5	0.01107	120.9	1005.7	1126.5
5	162.28	73.33	0.01364	130.1	1000.3	1130.5
10	193.22	38.38	0.02606	161.1	982.0	1143.1
14.7	212.00	26.79	0.03732	180.0	970.4	1150.4
20	228.00	20.08	0.04980	196.1	960.0	1156.2
25	240.10	16.30	0.0614	208.4	952.0	1160.4
30	250.30	13.74	0.0728	218.8	945.1	1163.9
35	259.3	11.89	0.0841	227.9	938.9	1166.8
40	267.3	10.49	0.0953	236.1	933.3	1169.4
45	274.5	9.39	0.1065	243.4	928.2	1171.6
50	281.0	8.51	0.1175	250.1	923.5	1173.6
55	287.1	7.78	0.1285	256.3	919.0	1175.4
60	292.7	7.17	0.1394	262.1	914.9	1177.0
65	298.0	6.65	0.1503	267.5	911.0	1178.5
70	302.9	6.20	0.1612	272.6	907.2	1179.8
75	307.6	5.81	0.1721	277.4	903.7	1181.1
80	312.0	5.47	0.1829	282.0	900.3	1182.3
85	316.3	5.16	0.1937	286.3	897.1	1183.4
90	320.3	4.89	0.2044	290.5	893.9	1184.4
95	324.1	4.65	0.2151	294.5	890.9	1185.4
100	327.8	4.429	0.2258	298.3	888.0	1186.3
105	331.4	4.230	0.2365	302.0	885.2	1187.2
110	334.8	4.047	0.2472	305.5	882.5	1188.0
115	338.1	3.880	0.2577	309.0	879.8	1188.8
120	341.3	3.726	0.2683	312.3	877.2	1189.6
125	344.4	3.583	0.2791	315.5	874.7	1190.3
130	347.4	3.452	0.2897	318.6	872.3	1191.0
135	350.3	3.331	0.3002	321.7	869.9	1191.6
140	353.1	3.219	0.3107	324.6	876.6	1192.2
145	355.8	3.112	0.3213	327.4	865.4	1192.8
150	358.5	3.012	0.3320	330.2	863.2	1193.4
155	361.0	2.920	0.3425	332.9	861.0	1194.0
160	363.6	2.834	0.3529	335.6	858.8	1194.5
165	366.0	2.753	0.3633	338.2	856.8	1195.0
170	368.5	2.675	0.3738	340.7	854.7	1195.4
175	370.8	2.602	0.3843	343.2	852.7	1195.9
180	373.1	2.533	0.3948	345.6	850.8	1196.4
185	375.4	2.468	0.4052	348.0	848.8	1196.8

[1] Reproduced by permission from Marks and Davis's steam tables and diagrams (copyright, 1909, by Longmans, Green & Co.).

TABLE VII.—PROPERTIES OF SATURATED STEAM (*Continued*)

1 Pressure lb. per sq. in. P	2 Temp. degrees F	3 Vol. cu. ft. per lb. V or S	4 Weight, lb. per cu. ft. V	5 Heat of the liquid Q	6 Latent heat of evap. L or R	7 Total heat of steam H
190	377.6	2.406	0.4157	350.4	846.9	1197.3
195	379.8	2.346	0.4262	352.7	845.0	1197.7
200	381.9	2.290	0.437	354.9	843.2	1198.1
205	384.0	2.237	0.447	357.1	841.4	1198.5
210	386.0	2.187	0.457	358.2	839.6	1198.8
215	388.0	2.138	0.468	361.4	837.9	1199.2
220	389.9	2.091	0.478	363.4	836.2	1199.6
225	391.9	2.046	0.489	365.5	834.4	1199.9
230	393.8	2.004	0.499	367.5	832.8	1200.2
235	395.6	1.964	0.509	369.4	831.1	1200.6
240	397.4	1.924	0.520	371.4	829.5	1200.9
245	399.3	1.887	0.530	373.3	827.9	1201.2
250	401.1	1.850	0.541	375.2	826.3	1201.5

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